Heating · Piping and Air Conditioning

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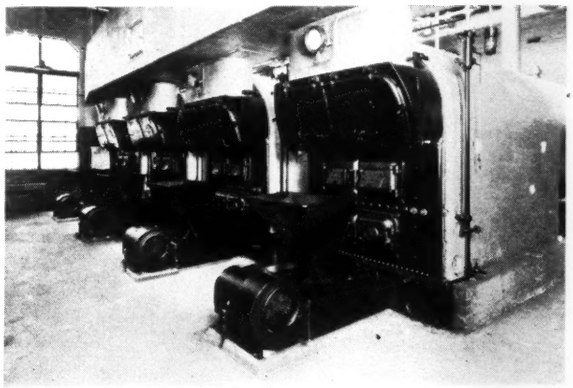
[Selection · · · Application · · · Operation]

STOKER FIRING

Low-Pressure Heating Plants

By **Clint E. Beery**

Four motor-driven mechanically-operated plunger-feed side-cleaning stokers with firebox boilers in a warehouse installation



THE origin of the underfeed stoker, both worm- or screw-feed and plunger or ram types, dates back some three-score years, coincident with the beginning of a general use of steam power.

This fact, however, does not signify that every underfeed stoker offered on the market today is the product of 60 years' successful development. The relatively few manufacturers who have through the intervening years built up a successful and permanent enterprise in marketing this type of stoker are those who have uniformly treated its installation as an engineering problem. Also most of their jobs have been of intermediate size, smaller than the great power jobs, and larger than the average low-pressure heating system. This group today treads with caution where the less experienced rush forward with a whoop.

Today heating and air conditioning are recognized as major items in the construction and operation of commercial buildings, yet only within the past four or five years have improved methods of burning coal been made available for general use in the smaller low-pressure heating system. The necessity for operating economy

TO THE ENGINEER and contractor responsible for design, installation, operation and maintenance of low-pressure heating plants for buildings, the subject of automatic coal burning deserves careful study. There are thousands of existing buildings wherein improved heating and great economies can be effected by the use of well-designed stokers, properly installed and intelligently operated. The stoker-fired low-pressure plant has its own peculiar problems that must be understood if the best results are to be obtained. These are presented in this article by Mr. Beery, long closely associated with the boiler and stoker industries.

now makes automatic coal-burning equipment a needed adjunct to the mechanical equipment of modern buildings. Likewise there are tens of thousands of existing buildings wherein improved heating and great economies can be effected by the use of well-designed stokers, properly installed and intelligently operated. To the engineer, architect and all who share responsibility of design, maintenance and operation of buildings the subject of automatic coal burning deserves more than the casual thumbing of stoker illustrations. These too often originate where sales appeal is the thought, instead of basic engineering data and their practical application.

Stoker Industry a Husky Infant

Five years ago not more than a dozen firms were active or had attracted much attention in the manufacture and sale of automatic coal-burning equipment in the low-pressure heating field. By the end of 1932 trade papers listed 86 such, and at the close of 1933 this number had been increased to approximately 180 manufacturers of domestic, commercial and industrial-type stokers who were offering their young hopefuls to the public. Certain it is that many more will enter the list before the peak is reached and the wave begins to recede.

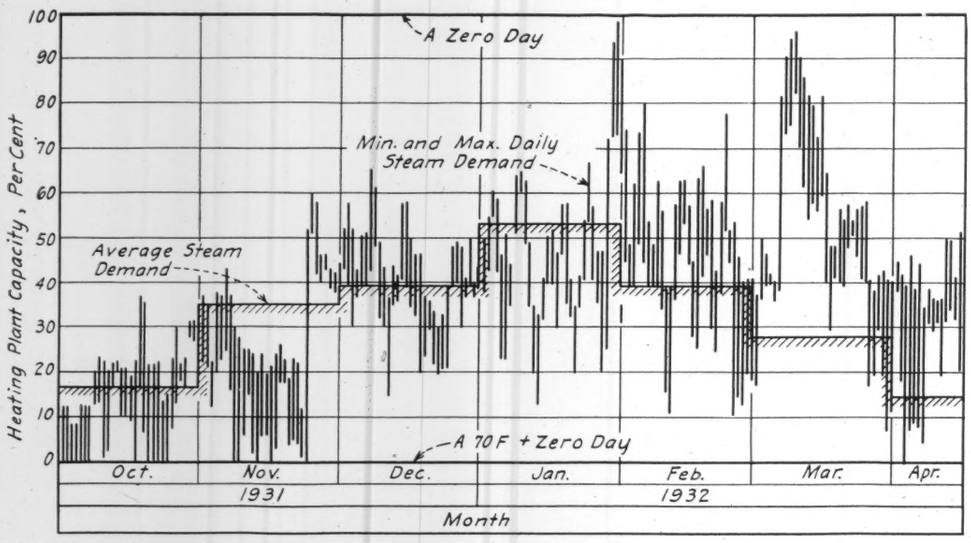


Fig. 1—Chart showing percentage of heating-plant capacity required to maintain 70 F inside temperature, for Chicago, 1931-1932

In the above chart the percentage of heating plant capacity required to maintain 70 F is shown for each day of the months indicated. The vertical lines extend from the point of minimum to the point of maximum steam demand on each day, based upon official temperature records for Chicago, 1931-1932. The bottom or base line indicates 70 F outside, no steam demand. The top line indicates a zero day, or steam demand of 100 per cent, equalling ¼ lb per sq ft of radiation per hr and heat emission of 240 Btu per sq ft per hr.

The step curve shows the daily average steam demand for each month to supply the degree day deficiency, in per cent of the plant's capacity, assuming the boiler and stoker to have 30 per cent excess, or reserve, capacity beyond the zero day demand of ¼ lb of steam per sq ft. For example, the average heating load for the month of January was 53 per cent of plant capacity.

Coal a Basic Industry

Coal is one of the most important assets of the United States. Unless treated as such, it becomes a great liability, as it forms the basis of one of our greatest industries.

There are more than 5000 coal mines in 26 states of the union. Mining activity alone supports a million and a half people, and a like number are occupied with its sale and distribution. In Illinois alone, some 45,000 people, exclusive of mining operations, are dependent upon coal for a livelihood. Directly and indirectly more than five million people of the United States depend wholly or in considerable measure upon the welfare of the coal industry. Approximately 20 per cent of the annual tonnage is used for domestic heating. Industries use 20 per cent and the railroads 25 per cent.

It should interest the reader to know how the coal resources of the world are divided among the nations:

United States	52 %	England	2.6%
Canada	16.6%	Siberia	2.3%
China	13.5%	Oceania	2.3%
Germany	5.7%	Asia	1.5%

Other countries each have less than 1 per cent of the world's supply.

While research and statistical records are incomplete and inadequate at this time, the following figures are substantially true and correct and show the remarkable growth of this infant industry during a period when old established business was struggling to exist.

In 1933, more than ten million dollars worth of stokers were built and sold in the United States. The potential market for commercial size stokers has been estimated at 1,500,000 units, or a manufacturers' gross volume amounting to $750,000,000, and these figures do not take into account repairs, replacements and new buildings to be constructed in the future. These figures are imposing and should be of considerable interest to architects, engineers and heating supply groups.

However, some confusion and disappointment may be expected during the years immediately ahead, for at this time too many manufacturers apparently think that there is a short cut to success in the stoker industry. Having created a mechanism that will feed coal into the furnace, they become flushed with enthusiasm and rush such a product to the market.

Many stokers designed largely upon library information and data secured through conventional avenues will

experience numerous unexpected disappointments that will require two or three seasons to analyze and correct. Where manufacturers lack patience, foresight and financial stability their success will be retarded proportionately. Nowhere does the trite old saying, "there is no substitute for experience," apply more forcibly than in the problem of designing a stoker that will overcome the thousands of unforeseen variables, in coal, plant, and habits of operation.

The first real fundamental upon which to build permanence is the result obtained in the furnace or firebox of a boiler. Innumerable designs in drive mechanism may be effective and have attractive sales points of advantage as such. Yet, however wide variations may be, all who attain satisfactory furnace performance will be employing quite identical stoker retort and furnace design.

High Pressure and Low Pressure

There are thousands of low-pressure heating systems which are legitimate prospects for stoker sales, where there is but one such working at high pressure. Therefore, low-pressure practice should command our careful study. The consulting engineer and manufacturer must ultimately understand that there is a difference, and recognize this difference in his practice, or oft be confronted with confusion.

Simply to change the control devices on a highly successful low-pressure operation may not result in an equally satisfactory job when operated at high pressure. By the same token, a successful high-pressure operation cannot be changed to low pressure by a change of control devices only, with absolute assurance of equally satisfactory results.

There are several reasons for this, prominent among which are two items of utmost importance—first, type and design of boiler employed; second, variation in load demand and its relation to plant capacity.

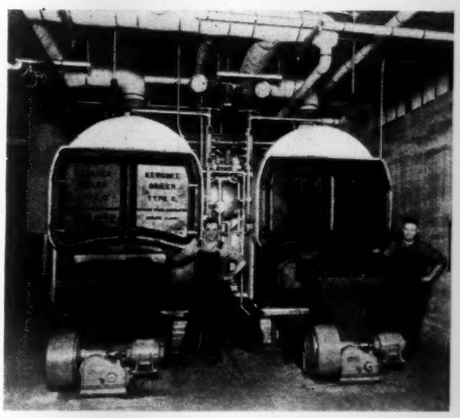
By the term "high pressure" as employed in this discussion we refer to operations where steam demand, steam pressure and fuel rates, once determined, remain fairly constant for extended periods of time; and where steam temperatures approximate 335 F or higher. Here a much higher furnace temperature is allowable, one that is more consistent with CO_2 determination. Here uniform load demand forms the basis for an equation upon which proper apportionment of all major factors of the problem can be determined.

In low-pressure heating there is no similar basis for such an equation. Here the whole problem is based upon a process of supplying latent heat to meet a widely varying demand. Here it is simply a matter of boiling water and being sure that conditions are favorable to its vaporization. Here 95 per cent of the fuel energy is invested in latent heat at low steam temperatures which seldom exceed 220 F. Here moderate furnace temperatures are more conducive to operating economy. Here provision for widely varying demand is essential, such variations often ranging from 150 per cent to as low as 15 per cent within a few hours. This is especially true where buildings are not heated throughout the 24-hour day. Buildings that are allowed to cool at night impose a heavy demand upon the boiler in the morning, and both boiler and stoker should have at least 30 per cent excess capacity above normal demand for heating-up periods and extreme weather conditions.

In stoker-operated low-pressure heating systems, the necessary flexibility is best provided for through a series of fuel rate changes, with a manually-operated air adjustment suited to each cycle or change of fuel rate; with control devices set to start and stop operation automatically, maintaining the desired steam pressure. While it is natural for combustion engineers to think in terms of combustion efficiency, it must be remembered that in low-pressure heating, seldom, and then but for short periods, does the demand for steam make it possible to run furnace temperatures consistent with high combustion efficiency.

For Chicago and vicinity (as an example), the average range of the heating load based upon the percentage of steam needed to heat to 70 F on a zero day equals approximately 44 per cent of the normal maximum. Where a boiler is stokered to operate at a minimum of 30 per cent overload capacity, for extreme weather and heating-up periods, the average heating load necessary to supply the average degree day deficiency for each month of the heating season approximates 32.7 per cent of plant capacity. Obviously, then, heating economy is a more consistent objective than combustion efficiency, where used in relation to low-pressure heating practice. See Fig. 1.

From the standpoint of satisfactory heating and economical operation, the experienced heating engineer is fully as important as the combustion engineer. The



Two screw-feed stokers in low-pressure heating plant in a factory

combustion engineer all too often clings to text-book data and orthodox theories that apply only where high temperatures are appropriate. With his chin up, he fails to follow the problem into the valleys and over the hills. His impulse is to provide highly efficient combustion, capable of producing several times the steam needed, and then say, "bring on the rest of the factors, surround my efforts with them, then note my success." All of which reminds the writer of the story of a tramp who promised the farmer's wife to kill all the rats on the farm in exchange for a good dinner, and after eating his fill, instructed her to bring on the rats.

Stokers and Boiler Design

Because the design and application of automatic coal-burning equipment is so intimately associated with good and bad boiler design, it justifies some mention of the boiler manufacturers' sins of commission and omission. The heating engineer, when aligned with the stoker manufacturer, must eventually result in better and more standard designs of heating boilers. Boiler manufacturers have often been more diligent in following commercial and competitive trends than in establishing correct engineering principles and standards in heating-boiler design. Fig. 2 constructed to scale from several

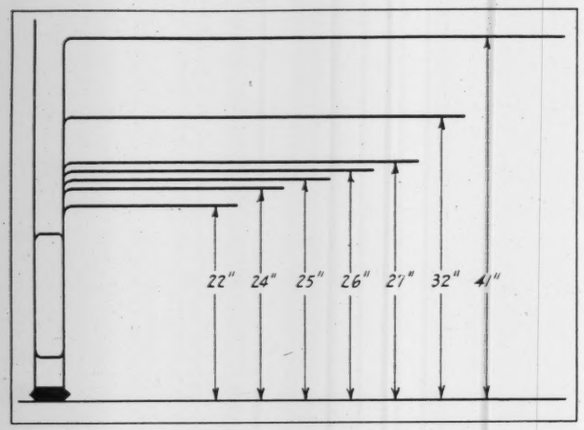


Fig. 2—Furnace height variation for same coal rate for seven well-known, widely-used steel heating boilers, approximate rating 4000 sq ft each

manufacturers' catalog data, discloses that seven well-known and widely-used steel heating boilers, of approximately 4000 sq ft rating each, all designed to burn the same amount of coal hourly, show a variation of nearly 100 per cent in their respective furnace heights. Little justification there for the boiler manufacturer to 'high-hat" the stoker industry for lack of engineering standards.

Furnace Design of Utmost Importance

The major factors of good combustion reduced to simple language are—carbon, oxygen, time, mixture and temperature. It is not our purpose to rehash scientific theories or combustion formulas except insofar as they emphasize the importance of proper furnace volume and dimension. In general, the most practical results will obtain when and where approximately 225 cu ft of air are being supplied to each pound of coal burned hourly, ⅔ of which supports combustion and ⅓ represents 50 per cent excess air, and the general conditions are normal to such apportionment.

Air being invisible, its volume makes no appeal to the imagination unless made tangible by illustration and dimension of the space it fills. To provide such an example, assume on a zero day we make steam to supply 1000 sq ft of radiation, which in turn maintains 70 F inside temperature. If we use coal with 12,000 Btu per lb content, we must burn approximately 35 lb per hr at 60 per cent efficiency. To do this we must supply the furnace with about 130 cfm of air. This heated to 1900 F expands to about 4½ times its original volume, giving nearly 590 cu ft of hot gases. If the boiler absorbs the heat properly the gases will cool to about 450 F before being ready to pass out through the smoke pipe and up the stack; at which temperature the gases will approximate twice the volume of the entering air, and require the exhaust of about 260 cu ft of gaseous mixture for each minute of operation. Expecting success on any other basis than properly proportioned furnace volume is as foolish as trying to catch elephants with mouse traps.

Where other conditions are normal, the furnace dimensions in the following tabulation will be found satisfactory in low-pressure heating practice:

Size of Stoker in Lb Coal per Hr	Desirable Furnace Volume in Cu Ft	Smallest Furnace That Will Serve, Cu Ft	Desirable Furnace Height, in In.	Lowest Furnace Height Safe to Use, In.
30	10	7.5	30	26
60	20	15.0	32	28
90	30	22.0	34	30
125	41	31.0	36	32
225	75	56.0	40	34
350	116	87.0	44	38
500	166	125.0	48	42
800	265	195.0	54	46
1000	333	250.0	60	54
1200	400	300.0	70	60

Another rule frequently used is to provide 1 cu ft of furnace volume to every 50,000 Btu heat release. Stated another way this approximates 1 cu ft to each 4 lb of coal burned hourly. However, many consider it better practice to provide 1 cu ft to each 2 lb of coal burned. Good understanding of the proposed operation is essential to selection of appropriate practice.

The Stoker as an Investment

Many glamorous claims are made showing enormous savings after the installation of stokers. We have no desire to challenge such claims but for ease of comprehension will assemble a representative group of self-supporting and provable facts into a composite picture (Table 1). This affords a bird's-eye view of the average probabilities, and will serve as a basis by which to appraise extreme and isolated examples of unusual savings. By a tabulation and grouping of various size jobs, from a cottage to an apartment hotel with 18,000 sq ft of radiation, we have a hypothetical example, mighty close to actual facts as a check of existing buildings will disclose.

In Table 1, Pocahontas coal at 14,000 Btu and 40 per cent efficiency is compared with stoker coal from midwestern states, at 12,000 Btu and 60 per cent efficiency. In taking these efficiencies it should be borne in mind that jobs vary so widely in conditions that a general assumption of 40 and 60 per cent efficiencies respectively is, in the writer's experience, practical but subject to qualification. Many claims of higher efficiencies than these can be substantiated just as many fail of fulfilment.

In hand-fired jobs burning Pocahontas coal, because this coal releases heat at comparatively slow rates of burning, it is safe to estimate that such jobs will often

operate around 45 per cent efficiency. The same job hand-fired using midwestern free-burning coal will seldom average better than 30 per cent efficiency, because such coal requires higher rates of burning for effective heat release, and unless given careful attention by the operator will burn through, leaving holes in the fire and large volumes of excess air to cool the furnace gases.

Again, for purposes of comparison and use of stoker engineering data, it has become general practice to "assume" an operating efficiency of 65 per cent. However, this assumption leaves the gate wide open to disappointment and possible controversy. This assumed efficiency is often too high for average operating conditions, which are beyond control of the stoker manufacturer or of the operator on the job.

The average steel heating boiler, stoker-fired, works at its highest efficiency when delivering approximately 70 per cent to 80 per cent of its normal capacity. Where the average heating load approximates only about 33 per cent of the boiler's capacity, it will be seen that in theory only does combustion efficiency mean boiler efficiency and operating economy.

Combustion efficiency requires reasonably high furnace temperatures, and there are few steel heating boilers that will not let heat escape up the stack, or radiate it to the boiler room as furnace temperatures begin to exceed 2000 F, resulting in a corresponding drop in operating efficiency. In low pressure heating, except in the Northern cold states, only at occasional intervals and for short periods are conditions favorable to high over-all operating efficiency.

The Screw-Feed and the Ram-Type

In selecting a stoker for use on a specific job, the type and setting of the boiler, the size and character of the load and the general conditions surrounding the operation are more important for first consideration than the type of stoker to be used. Where capacity and furnace performance are equal, there is little to merit any choice between the screw-feed and the ram types from the owner's standpoint.

On the smaller jobs where there is but one fire door on the boiler, the screw-feed type can be installed more conveniently because the feed mechanism and coal hopper can be placed at one side, at an angle, or in front, always at some distance from the fire door, allowing ample room for cleaning and stoking fires. Where it is necessary to install the ram-type stoker in front of such a boiler, the hopper must be emptied and removed to give access to the fire, or else the stoker must be pitted low enough to permit the use of special fire-attention doors below the mud ring, or bottom of the water leg of the boiler.

This in turn requires extra refractories to wall up the furnace to the water-protected heating surface. In the case of smaller boilers and narrow furnaces, this often results in bottled heat in the furnace. Instead of the heat being absorbed by the boiler and put to useful work, it is stored up in the burning fuel, re-fusing the ash and causing much unnecessary clinker distress, often requiring a search for coals with a high ash fusion temperature to keep the job running satisfactorily.

This condition tapers off and becomes much less acute as boilers and heating loads increase in size. Individual characteristics of a job should always be the guide in selecting the type of stoker appropriate to the specific operation.

On intermediate size jobs, there is but little to influence any choice between the screw-feed and the ram-type stokers, except adaptability to existing conditions, or where cost and convenience make re-arrangement desirable.

On larger units where coal-burning rates begin to exceed 1200 lb of coal per hr, another phase of the problem becomes apparent; namely, proper distribution and control of the fuel as it comes into the burning zone. Here the ram type can continue the use of a reasonably narrow retort, and with the aid of adjustable auxiliary pushers level and control the coal as it enters the burning zone.

Coal for Automatic Coal Burners

Perhaps no substance was ever so long and intimately associated with human welfare and yet so little understood as coal. To the economist it offers intriguing opportunity for study and discovery of new uses. To the

Table 1—The Stoker as an Investment

Type of Building	Tons Yearly	Sq Ft Radiation	Yearly Fuel Cost	Yearly Saving	Saving Per Cent	Return on Investment Per Cent	Minimum Stoker Lb per Hr
Residence		400					
Pocahontas	15.4		$115				
Stoker	14		73	$ 42	37	24	30
3-Flat		900					
Pocahontas	34.6		259				
Stoker	32.6		163	96	37	21	60
6-Flat		1800					
Pocahontas	69.3		520				
Stoker	65.2		326	194	37	30	125
9-Flat		2700					
Pocahontas	104.6		784				
Stoker	97.7		488	296	37	37	150
12-Flat		3600					
Pocahontas	138.6		1039				
Stoker	130.3		651	388	37	40	225
24-Flat		7200					
Pocahontas	277.2		2041				
Stoker	260.6		1303	738	37	67	350
36-Flat		10800					
Pocahontas	416		3120				
Stoker	391		1955	1165	37	86	500
Apt. Hotel		13000					
Pocahontas	500		3750				
Stoker	471		2355	1395	37	92	800
Apt. Hotel		15000					
Pocahontas	576		4320				
Stoker	543		2715	1605	37	95	1000
Apt. Hotel		18000					
Pocahontas	693		5197				
Stoker	651		3255	1942	37	103	1200

Note: Pocahontas figured at $7.50 per ton, no account taken of any probable price differential where tonnage runs higher as on the larger jobs.

practical-minded its appeal is natural. In the realm of romance and drama, no field is richer in human experience than the story of coal. However, space demands that we stick to such phases of coal as interest those who directly or indirectly have to do with its use in stokers and low-pressure heating.

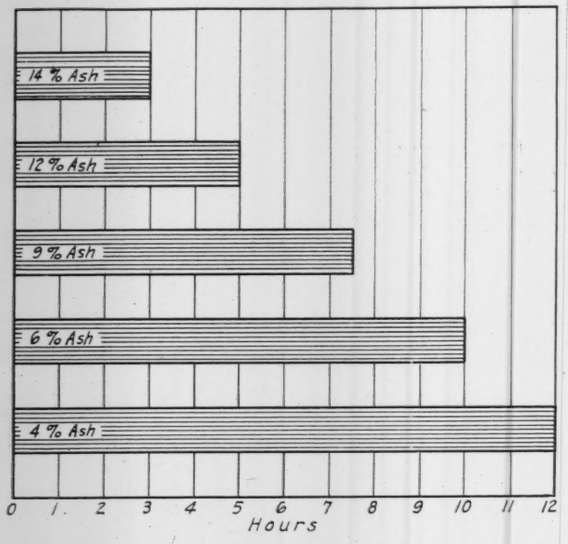


Fig. 3—Effect of varying ash percentage in coals on time fires will run before cleaning becomes necessary under the same load conditions

In this connection, for the sake of lucidity we will approach some such characteristics of coal by pointing out their extremes. One such is the coking *vs.* the free-burning characteristic. Pocahontas is perhaps the most widely known and extreme example of this coking, or "caking" tendency. This coal, as it becomes heated, swells, expands in volume, and fuses into a solid mass at very low temperatures. It will burn slowly and generate steam-producing temperatures with moderate furnace activity. It runs high in Btu and in carbon, but low in ash and volatile matter. It is very desirable coal for hand-firing, but is seldom used with underfeed stokers.

The other extreme is a coal that burns so freely that once the volatiles are distilled practically no bed of incandescent carbon remains. It extinguishes quickly and dissolves into a light feathery ash, and cannot be burned slowly with satisfactory furnace temperatures resulting. It is high in ash and volatiles and low in carbon but normal in Btu value in comparison to other midwestern coals. When used with underfeed stokers it requires a deeper fuel bed, and calls for frequent cleaning of fires. Often greater tonnage will be necessary as against other midwestern coals having a slight caking tendency.

Another important item is the amount of ash content in coals. This variation ranges from 2 per cent to 16 per cent, or in other words some coals have 8 times more ash than others. To understand the effect of high and low ash content in coal on stoker operation is to be forearmed when turning jobs over to inexperienced or indifferent firemen. Often this will keep a job sweet that otherwise might go sour because the new deal does not fit into the past experience or habits of the operator. See Fig. 3, which shows the effect of varying ash percentage in coal on the time fires should run between cleaning periods.

To prevent confusion, we refer to coal analysis but little. Coal analysis, as generally understood and used, provides no key to the burning characteristics of coal. In other words, burning characteristics, and not coal analysis, determine what coal is best suited for use with underfeed stokers.

Fly-Ash and Its Correction

One fly in the ointment of some manufacturers, just as they are about to shout Eureka!, is the problem of fly-ash. For the most part, fly-ash problems are due to one of three things, or combinations of them.

They are, though not in the order of their importance: --(1) too high an air velocity through the fire; (2) inadequate furnace height and volume; (3) excessive draft. Some coals aggravate the situation, but to change coal without correcting the above errors where they exist will not result in permanent improvement and satisfaction.

Excessive draft can be brought under control by the use of any one of the several approved balanced-draft equipments available. Where careful check on economy is desired, the use of such as regular boiler-room equipment is commendable.

The other two causes are often found jointly contributing to the difficulty. Too high an air velocity through the fire paralyzes the time interval necessary to combustion and carries particles of unburned fuel matter up out of the fire-bed. These particles cannot be burned in suspension without temperature and time to complete combustion. Where furnace heights are too low and proper furnace volume is lacking, these particles impinge against metal heating surfaces much lower in temperature than the fire. This cools them, adds to the difficulty, forming a gritty sand-like substance containing a large amount of unburned carbon, though the volatiles may be completely burned and the furnace operating smokelessly.

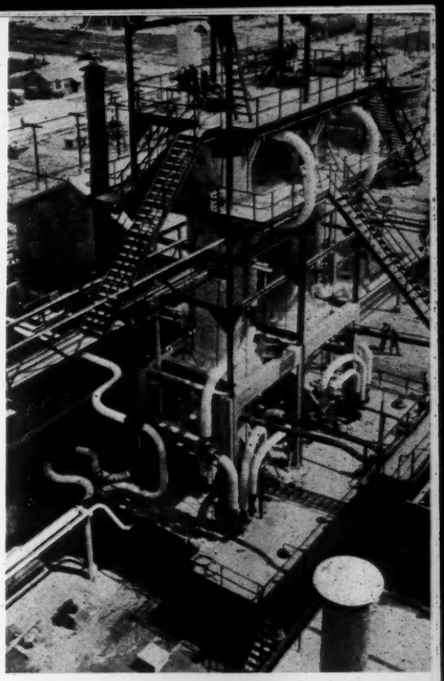
The problem of fly-ash *must be solved*. Manufacturers who disregard this will ultimately meet determined resistance from smoke-abatement departments, health agencies and anti-air-pollution activities in all of our larger cities.

Smoke Abatement

Mass education is a plant of slow growth. One of our greatest phenomena in American life is the perpetual waste of our resources. All important cities now realize not alone the benefit but the necessity for effective smoke-abatement departments. No resident of any major city for the past 20 years can deny the wonderful results of such activity, nor estimate the damage done in the past by the great smoke palls that daily hovered over our cities. The job, however, is far from complete. Education, practical information, improved equipment, patience and perhaps a big stick will be necessary for some time to come.

Oil Piping Systems

for Production, Transmission and Refining of Petroleum



The New Code for Pressure Piping

By **Sabin Crocker***

REQUIREMENTS for oil piping, materials, thickness of wrought pipe, and thickness of cast-iron pipe as outlined in Section 3 on oil-piping systems in the new Code for Pressure Piping were outlined and discussed in the writer's article last month. Among the tables and charts presented with that article was included a summary of requirements and recommendations for oil-piping systems.

The present article discusses a number of other requirements of this section of the Code.

Valves

Comprehensive requirements are included to govern design, materials, and use of valves, as quoted below.

"Valves. (a) All valves, wherever practicable and except as noted below, 2½-inch in size and larger shall be of the outside screw rising-stem type. Under 2½-inch, the valves may be of the inside-screw rising-stem type. Drilling valves and valves used on lines containing acid shall be preferably of the inside-screw non-rising-stem type. Class 11[1] valves may be either of the rising-stem or non rising-stem type. Valves to be operated under conditions where it is unnecessary to know whether they are open or closed from a safety standpoint may be either of rising or non-rising-stem type. It is recommended that all non-rising-stem valves, except drilling valves and valves smaller than 2½ in., be equipped with indicators to show position of gates or discs.

"(b) Every valve shall be plainly marked with the maker's name or trade-mark and the rating, so that the working oil pressure and temperature for which it is guaranteed may be determined from the manufacturer's catalog or data sheet.

"(c) Valves covered by Classes 1 and 2 in sizes 4 in. and larger shall have bolted bonnets. Valves smaller than 4 in. may have screw bonnets, preferably of the union type. It is recommended that valves 4 in. and larger have flanged ends.

"(d) Valves under Classes 3 to 11, inclusive, 2½ in. and larger, shall have bolted bonnets. Valves under these classes smaller than 2½ in., may have screw bonnets of the union type, but it is recommended that 1½-in. valves and larger under these classes have bolted bonnets. Valves under these classes, 2½-in. and larger, shall have flanged ends when used at temperatures above 450 F.

"(e) Brass or bronze trimming or yoke bushings shall not be used in valves where the operating temperature is more than 400 F. Bushings and trim may be of an alloy steel, or other material with a high melting point."

"Check Valves. Ball-check valves of the straight-through type are recommended for Classes 5 to 10 inclusive. When ball-check valves are used in horizontal positions, the balls shall be guided so as to move in as near a horizontal line as practicable. Ball-check valves with globe type bodies are not recommended for this service."

"Special Valves and Cocks. (a) All special valves, such as cocks, relief valves, etc., shall conform to the material and strength specifications required for the class of service in which they are used.

"(b) For high-temperature service, where cocks are used, they shall be so designed as to prevent galling, either by making the plugs of a different material from the body of the cock, or by treating the plug to insure different physical properties. Where cocks are used for services covered by Classes 3 to 10, inclusive, special consideration shall be given to their design, so as to insure that they will be readily operated under hot-oil conditions. Cocks which are not gear- or worm-operated shall either have their operating levers attached to the stem or the levers shall be maintained in locations readily accessible for emergency operation."

"Relief Valves and Automatic Control Valves. (a) All automatic-control valves shall be bypassed or otherwise installed in the line so as to permit manual operation in the event of failure of the automatic valves. Relief valves on equipment where service pressure is more than 7½ lb shall be connected by piping so as to discharge at places where the danger from flashing will be a minimum.

"(b) Where equipment, within which pressure can be generated because of service conditions, is protected by a relief valve, there shall be no stop valves permitted between the vessel and its relief valve, except for inspection or repair purposes as permitted in (*d*) and except in cases where multiple relief valves are provided and the stop valves are so constructed that they cannot be operated so as to reduce the pressure relieving capacity below that required.

"(c) Where equipment, within which pressure originates from an outside source exclusively, is protected by a relief valve, there may be stop valves between the vessel and its relief valve which need not be locked open if the stop valve also closes the vessel from its source of pressure. Any stop valve between the relief valve and the vessel which does not close the vessel from its source of pressure shall meet the requirements given in (*b*) above.

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*Engineer, The Detroit Edison Company, and Member of Board of Consulting and Contributing Editors. Among other standardization committee affiliations, Mr. Crocker is Chairman of Subcommittee *1* on Plan, Scope, and Editing of A. S. A. Sectional Committee *B31* on Code for Pressure Piping.

This article continues the series on piping standardization, which started in the September, 1933, H. P. & A. C.

[1]See Table 2, p. 104, March, 1934, H. P. & A. C.

"(d) Where equipment, which is normally in operation for indefinite periods at temperatures below 200 F, is protected by a relief valve, it may have a stop valve between it and its relief valve for use in inspection and repair, in which case the stop valve shall be arranged so that it can be locked open at all times, except for short periods when it is necessary to remove the safety valve for purpose of repair.

"(e) Stop valves, as permitted above, shall be gate valves, or globe valves installed so that the disc will open against the pressure, except that this limitation need not apply to stop valves closing the vessel from its source of pressure."

"Double Valving and Drips and Drains. (a) Double stop or cut-off valves shall be employed in all oil, oil vapor or refinery gas lines inter-connecting equipment normally operating at temperatures of 150 F or over, or at a pressure of 1 lb gage or more where equipment has to be entered by men for cleaning or repairing. As an alternate, the line entering the equipment to be cleaned or repaired shall be blanked. When double stop valves are used, one of the valves shall be placed as close as possible to the equipment to which the pipe connects so as to preclude any pipe or nipple between this valve and the equipment.

"(b) Suitable drips or drains shall be installed between these double stop valves. The minimum size of drip connections shall be ¾-in. The valves and fittings used on drip and drain lines shall be of the same class as the valves and fittings on the main line to which the 'drip' connects.

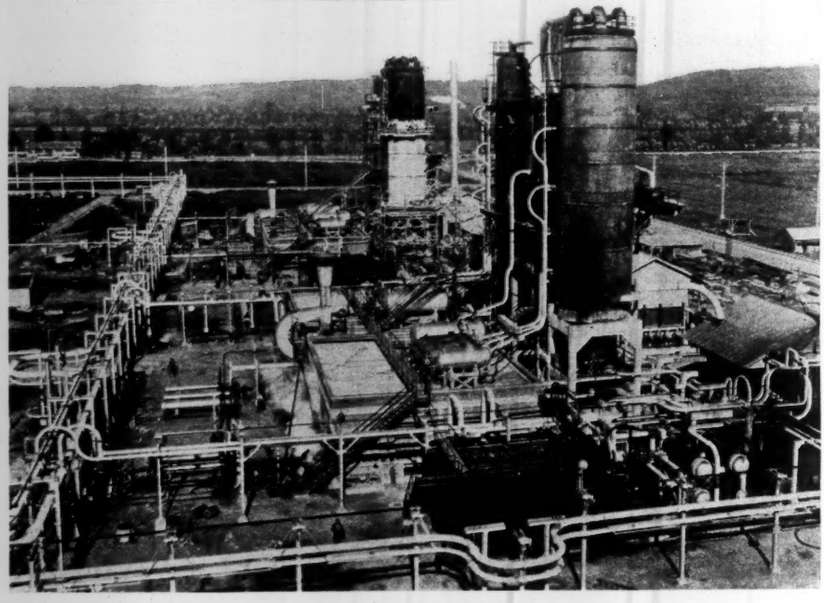
Note: The above requirement is not intended to apply to lines connected to 'oil tankage.'"

Table 3—Recommended Maximum Nominal Sizes in Inches for Cast or Forged Steel Screwed Fittings and Flanges

	Service Pressure in Lb per Sq In.						
Temp F	1500	1000	900	600	400	250	100
950	¾	1	1	1½	2	2½	3
900	¾	1	1½	2	2	3	4
600	2	2	2	3	3	4	6
400	3	3	3	4	4	6	8
300	3	3	4	4	6	8	10
200	3	4	4	6	8	10	12
Atmospheric	4	6	6	10	12	14	16

Table 4—Recommended Maximum Nominal Sizes in Inches for Cast or Malleable Iron Screwed Fittings and Flanges

	Service Pressures in Lb per Sq In.						
Temp F	1500	1000	800	600	400	250	100
400	2½	2½	3	3	3	6	6
300	2½	2½	3	4	6	6	8
Atmospheric	4	6	8	10	12	14	16



Joints, Flanges, and Fittings

The use of dimensional standards equal or superior to those called for in Table 2[2] is recommended. Where special fittings are required to overall dimensions differing from American Standard, they must correspond to the respective standard as regards wall thickness and flange dimensions. Recommendations concerning the use of screwed and flanged joints are set forth in the following paragraph entitled *"Fittings."*

"Fittings. It is recommended that within refinery limits the use of screwed fittings and screwed flanges for oil or refinery gas lines be limited to the maximum nominal sizes given in Tables 3 and 4. For conditions coming between those given in Tables 3 and 4, interpolations may be made. Screwed joints for temperatures above 400 F should be seal welded."

Further requirements concerning bolting, pipe threads, joints and gaskets are set forth as follows:

"Bolting. (a) Commercial machine bolts may be used to make flange connections for Classes 1, 2, 3 and 11. Connections in all other classes should have bolting in accordance with *A.S.T.M. A 96*, Class "B," or the equivalent. Alloy-steel bolts and stud bolts shall be marked in such a way as to distinguish them from commercial bolts. Bolts or studs shall extend completely through the nuts.

"(b) For conditions where bolts made of *A.S.T.M. A 96* material are required and for temperatures where "creep" will not be encountered, it is recommended that the bolts be either threaded their entire length or the unthreaded portion be turned down to the root diameter."

"Pipe Threads. All pipe threads on pipe shall be in accordance with the specifications for pipe threads given in the latest revision of the *American Petroleum Institute* specifications for line pipe (*A.P.I. 5L*) or the American Standard Pipe Thread (*A.S.A. B2*). Threads on valves and fittings shall be suitable for use with the pipe threads above mentioned. See also *Section 5, Chapter 2* of this Code.

"Pipe Joints. (a) Flanges shall be attached to pipe by threading, expanding, lapping (Van Stone), welding, or other approved methods described in *Section 5* of this Code.

"(b) Where threaded flanges are used in piping which is to be operated under the conditions covered by Classes 3 to 10, inclusive, for sizes 2 in. and larger the bore of the flanges may be beveled at the flange face and the pipe flared by 'peening' or rolling to fit the bevel. The hub of the flange for all sizes Classes 6 to 10, shall be seal-welded to the pipe, the welding covering all exposed pipe threads.

"(c) Lap (Van Stone) or similar joints with either smooth or serrated faces, are permissible for Classes 1 to 7, inclusive, and for Class 11. A ground-ball-type joint or an upset type of lap (Van Stone) or similar joint, which permits machining out a groove suitable for a ring or tongue and groove joint with a thickness equivalent to the full thickness

[2]See p. 104, March, 1934, H. P. & A. C.

The decorative illustrations accompanying this article are typical views of the sort of piping covered by the Code

of the pipe at the bottom of the groove, may be used for all classes. Forge and hammer, fusion, resistance, or other approved welded flange may be used for all classes.

"(d) Flanges for piping specified under Classes 7 and 8 may be faced for tongue and groove, male and female, ring type, or the ground-ball type of joint. Flanges for piping specified under Classes 9 and 10 shall be faced for either tongue and groove or ring type joints, or for the ground-ball-type joint.

"(e) Flanges for piping Classes 2 to 6, inclusive, and for Class 11 may have standard raised faces. Flanges for piping under Class 1 may have smooth faces.

"(f) Flanges used at the ends of pipe bends which are subjected to the strains of expansion and contraction shall not be lighter than the 250-lb American Cast-Iron Standard *(A.S.A. B16b)* or the 150-lb American Steel Standard *(A.S.A. B16e)*, even where on other parts of the line Class 1 fittings are permissible.

"Gasket Material. (a) Gasket material for piping under Classes 1 to 6, inclusive, if used for a temperature of over 250 F or if adjacent to a hot-oil line shall be metallic, asbestos or other non-burning material. Compressed asbestos gaskets are not recommended for temperatures above 750 F. Gaskets specified above shall be as thin as the finish of surfaces will permit to reduce probability of blowing out due to accidental increase of pressure.

"(b) Gaskets for piping under Classes 7 to 10 inclusive, except where ring joints or sargol or sarlun joints are used, shall be metal or metallic asbestos-filled gaskets, of such design and material that they will not be injuriously affected by the temperature conditions encountered.

"(c) Rings for ring-joint connections shall be of a metal of such composition that it will not be injuriously affected by the temperatures encountered. All gasket material shall be such as not to be attacked detrimentally by the fluid conveyed."

Miscellaneous Requirements

Several paragraphs on miscellaneous topics in which the subject matter is sufficiently different from other sections of the Code to warrant reproduction in full are quoted below.

"Thermal Expansion. (a) Allowance for expansion and contraction in high-pressure and high-temperature oil and oil-vapor lines shall preferably be made by the use of pipe bends whenever space permits. For changes in direction in high-pressure and temperature oil and oil-vapor lines, pipe bends are preferred to fittings whenever space conditions permit. Where construction conditions do not permit the use of pipe bends, expansion slip joints may be used except that for temperatures above 750 F the pressure shall be limited to 25 lb maximum. Corrugated expansion joints may be used where their design is suitable for the pressure and temperature conditions encountered.

"(b) Where pipe bends are used for expansion at temperatures not exceeding 750 F, and where the calculated expansion exceeds ¼ in., it is recommended, when practicable, that piping be cut short about 50 per cent of the amount of thermal expansion and cold sprung during erection.

"(c) Where expansion slip joints are used, they shall be slip guided with long stuffing boxes, and provided with stop limits to prevent excessive travel. The trim shall, in general, correspond to that of the valves used for the same class of service, except that no metal or alloy shall be used which, on account of high temperatures, is liable to "freeze" at the contact surfaces so as to prevent slipping. In general, dissimilar metals are recommended for sliding contact for this service.

"(d) Bellows type of expansion joint of flange steel plate construction may be used on vapor lines where the pressure is not over 25 lb.

"(e) For lower limits of temperature and pressure, covered by Class 1, swivel joints or swing joints made up of elbows and short lengths of pipe may be used. This type of expansion joint is not recommended for high temperatures and pressures.

"(f) For Class 11 construction, expansion and contraction may be provided for by laying the line with sufficient curvature or "slack." If expansion joints are used on underground pipe lines it is recommended that they be protected with suitable covering which will not interfere with their functioning as expansion joints."

"Pipe Covering. Where high-temperature oil or vapor lines are insulated, the insulation shall be protected with a fire-resisting covering material of proper mechanical strength to resist destruction by water from a fire hose."

"Proximity of Cold Oil Lines to Hot Lines. Where oil lines are in close proximity to lines or equipment containing hot products, so that the failure of valves and fittings on the oil lines would permit the oil to splash or come in contact with the lines or equipment containing hot products, thereby creating a fire hazard, cast- or forged-steel valves and fittings shall be used."

"Inspection and Tests. (a) Prior to erection, all material shall be designed and manufactured suitable for and capable of withstanding, without injury, the test requirements prescribed in Classes 1 to 11. Care shall be taken to remove all air from the interior before applying the test pressure. Steel castings, when tested, shall be struck several sharp blows with a machinist's hammer while under the specified pressure, in order to disclose hidden defects.

"(b) After erection, all assembled piping for service pressures exceeding 25 lb shall be suitable for and capable of withstanding, without injury or leakage, a hydrostatic or kerosene test pressure of not less than two times the service pressure.

"(c) Drilling valves shall be suitable for and capable of withstanding a hydrostatic test pressure at the place of manufacture equal to not less than twice the service pressure to which the valves are to be subjected.

"(d) The inside of all pipe, valves and fittings shall be smooth, clean and free from blisters, loose mill scale, and dirt when erected."

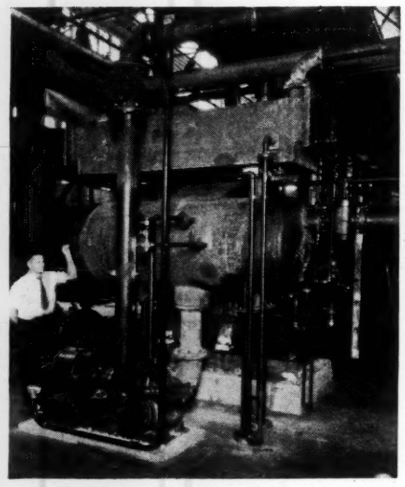
As is the case with other portions of the Code for Pressure Piping, the section on Oil Piping has undergone a progressive seasoning process applied through constructive criticism during the various steps of its formulation by *A. S. A. Sectional Committee B31.* The members of the Committee do not consider it perfect in the form reported in these articles; probably a number of imperfections will be brought to notice when the Code is presented to industry for review, and several are now in process of being ironed out before publication. For one thing the whole system of pressure-temperature rating classifications probably will be reduced eventually to a simple set-up based on data such as that shown in Figs. 1 and 2 last month, and in Table 1 shown on page 457 of the September, 1933, H. P. & A. C.

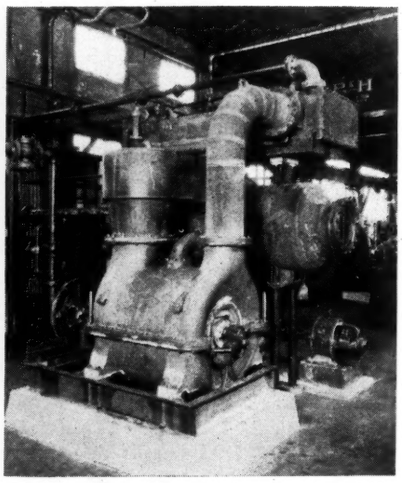
Another reason for the continuing need of change arises from rapid advances made in piping practice to accommodate higher pressures and temperatures, welded construction, new materials, and the like. After the Code is once established it will be desirable to set up some permanent form of organization similar to the *A. S. M. E.* Boiler Code Committee to interpret its provisions, and to effect revisions from time to time as found necessary.

[The following installment of this series of articles on Piping Standardization, concerns the sections of the Code for Pressure Piping dealing with Gas and Air Piping, District Heating Piping, and Fabrication Details.—*Editor.*]

Water-Vapor Refrigeration for Air Conditioning

—*the Centrifugal Compressor Unit*





Figs. 1 and 2—Experimental set-ups of centrifugal water-vapor refrigeration units

[*By* **Paul Bancel***]

WATER VAPOR as a refrigerant, while not a new idea, has enjoyed a rapid development of late, particularly as applied to air conditioning. As pointed out in an article in last month's issue, it depends upon compressing units capable of "sweeping" very large volumes of low-density vapor. These have taken two forms, the steam-jet booster and the centrifugal compressor, the development and characteristics of the latter being the subject of this article. It is of interest that a large motor-driven centrifugal machine is to be used in connection with the air conditioning of four floors in the RCA Building, New York City, which will be devoted to general office space. The machine will be installed on the 58th floor and will operate in connection with a cooling tower.

THE development of water-vapor refrigeration has depended upon the development of compressing units capable of "sweeping" very large volumes of low-density vapor, as was explained in the writer's article last month. These have taken two forms, the steam-jet booster (discussed in the March issue), and the centrifugal compressor, with which we are concerned in this article. In the centrifugal unit a centrifugal compressor of large capacity takes the place of the steam-jet boosters. The heat discharged to the condenser is therefore only the refrigeration load, plus the work of the compressor, as with the ordinary refrigerating cycle.

Development of the Centrifugal Machine

Two groups of engineers were primarily concerned with the development of this type of unit:—experts in centrifugal compression and experts in condensation and high vacuum. The turbo-compressor engineers had built hundreds of centrifugal machines for air and gases, for pressures as high as 110 lb, for capacities of over 100,000 cu ft, of one to fifteen stages, operating at 2,000 to 15,000 rpm. However, the problem in the present case could not be solved by the mere adaptation of one of these compressors.

An enormous volume of highly-attenuated[1] water

*Ingersoll-Rand Company, New York City. From a paper presented at the 26th annual meeting of the *American Institute of Chemical Engineers*, Dec. 13, 1933, Roanoke, Va. The first part of this paper, on steam-jet refrigeration, was published last month; the present installment concludes the paper.

[1]That is, a water vapor of low density.

vapor had to be compressed to a pressure compatible with ordinary condensing-water supplies. Since there would be no burden of steam load as with steam jets a somewhat better vacuum or lower back pressure could be expected with the centrifugal machine and the possible discharge pressures were set down as 1½ to 2 in. absolute (28 to 28½ in. vacuum). While these pressures might appear low, the ratio compared to a suction pressure of, say, 0.36 in. was surprisingly high—4½ to 6 times. This called for severe duty in the compressor.

In terms of a centrifugal compressor working with atmospheric intake, these ratios were equivalent to discharge pressures of 50 to 75 lb per sq in. gage. A compressor for 100 tons would need to have about 35,000 cfm capacity and such a machine were it built to operate with atmospheric air, discharging at 50 to 75 lb pressure, would require as many as 10 stages and 5,000 to 7,000 horsepower driving power. Manifestly such a machine would be entirely unsuited for compressing 35,000 cu ft of low-pressure vapor requiring less than 100 horsepower. Furthermore, it was not to be expected that highly-attenuated water vapor would behave in the same way as air or gases for which the design factors were well known. Considerably less compression could be expected with water vapor than with air. To add to the difficulty it was imperative that the machine be absolutely tight against air leakage in all joints and stuffing boxes. It is little wonder that the water-vapor cycle formerly had been considered a theoretical curiosity.

The condenser (while small as compared to ordinary power-house units) presented rather difficult and unusual problems. The discharge vapor would be superheated and condensation under these conditions was known to be difficult. A relatively high vacuum or low absolute pressure had to be maintained when using a minimum of condensing water and air removal power. The low water consumption called for sufficient equivalent tube length or number of passes so that the water would be heated as closely as possible to the vacuum temperature. The efficient removal of the unavoidable air leakage called for counter-current cooling to as near inlet condensing-water temperature as possible.

Enough has been said to sketch the complexity of the problem and the numerous difficulties confronted. The mere assembly of the component parts then available would be ridiculously impractical, costly, and inefficient. Each problem, however, was reconsidered in the light of the specific application, and the solutions developed combined into a full-size experimental plant. Some of the figures shown here illustrate the essential parts of the apparatus. The compressor was turbine driven, and of large capacity. Suction and discharge were at the top; the evaporator was cylindrical and fitted in various ways for studies of water distribution, separation, and removal. The condenser was above the evaporator and of the multi-pass counter-current type.

Extensive experiments were conducted. It was demonstrated that the cycle was entirely practical and the anticipated reactions to load variations proved. The result was a commercial machine. The compressor is set upon a base which also forms the evaporator and the condenser is attached to the compressor discharge, thus producing a compact and simple assembly. Standard sizes ranging from 50 to 300 tons are available. Instead of a motor with speed-up gear, a direct-drive turbine can be used. Speeds are from approximately 7,000 to 10,000 rpm, depending on size, which agree with recent practice in high-efficiency, small-turbine design.

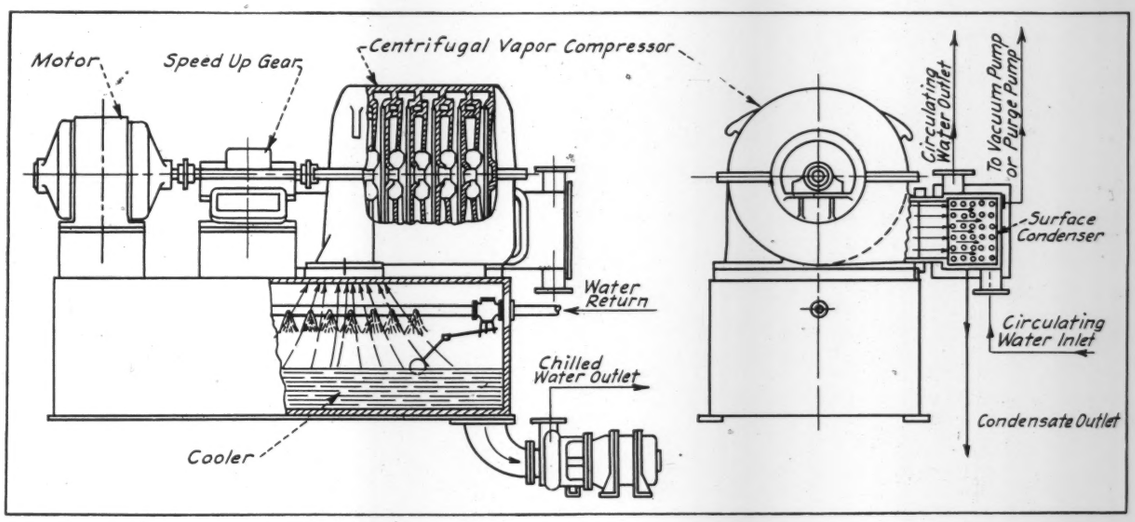


Fig. 3—Diagram showing operation of centrifugal water-vapor refrigeration unit

Horsepower Requirements

The theoretical power consumption for adiabatic compression of water vapor from 50 F to 86 F (the refrigeration ton standard) and alternatively to 95 F for example, are given below:

Suction Temp., F	Saturated Discharge Temperature	Adiabatic, Hp per Ton
50	86	0.371
50	95	0.476

The brake horsepower requirements of actual refrigerating units involve many factors of design and proportion; of these the efficiency of the compressor itself is probably most important, and there can be no question of the fact that a centrifugal compressor is usually less efficient than a reciprocating machine. This, however, is offset in this instance by the lower theoretical power

required. As a result the brake horsepower requirements of the centrifugal unit may be said to compare favorably with any type of refrigeration, frequently being less.

Operating Characteristics

The centrifugal water-vapor unit differs fundamentally from all other types in its part-load characteristics. In general, the centrifugal unit is operated at constant speed.

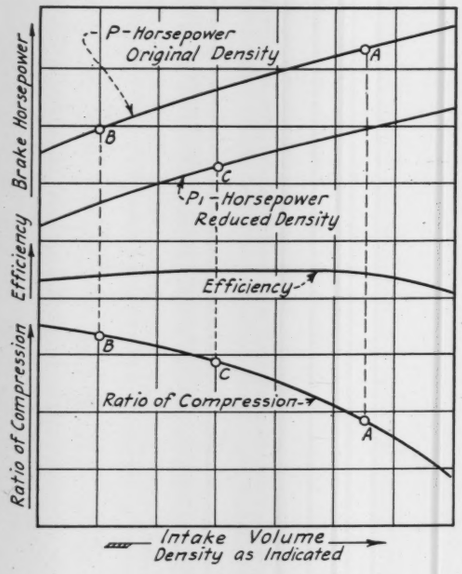


Fig. 4 — Operating characteristics of centrifugal vapor unit

Under this condition and without any other adjustments, reduced tonnage is automatically reflected in reduced power, approximately in proportion to the tonnage. The unit simply floats on the load. This feature has been fully verified in test and commercial operation. It follows from certain fundamental relations of evaporator, compressor, and condenser, and the inter-relation of their characteristics.

Fig. 4 shows a typical volume characteristic of a centrifugal compressor: As the volume decreases the ratio of compression increases, and over the designed range, there is a flat efficiency curve. The power is shown by line *P;* at part volumes the brake horsepower decreases but not proportionately. A second line P_1 below P shows the power for vapor of lower density. Within limits, a decrease in density at fixed volume results in an equivalent reduction in power. Now, with decreasing load, a change occurs in both volume and density but for simplicity they can be considered as if there were two distinct phases or steps.

1. First, a decrease in volume proportionate to the decrease in load as from *A* to *B*. The power then decreases along the curve *P* from point *A* to *B*.
2. Second, a decrease in density and increase in specific volume which now increases the total volume to intermediate value *C*. However, due to the lower density, the power is now determined by lower line P_1 and the absolute value at *C* is less than at *B*.

Decreased density at reduced load is the result of reduced suction pressure which, in turn, is due first to the greater ratio of compression which the compressor can generate at reduced volume. Secondly, in most cases the condenser pressure decreases with reduced load and as this is reflected at the suction of the compressor the evaporator pressure is further reduced with further reduction in density. In fact, there is frequently a cumulative effect because there is a tendency for the temperature of the condensing water to be colder at times of part loads, particularly in the case of small supplies or where cooling towers are installed. The three factors—the greater compression ratio available, the lighter heat load on the condenser and the colder condensing water, singly or combined—tend to reduce suction pressure and density.

The characteristics plotted against tonnage are shown in Fig. 5. Condensing water quantity is maintained constant for all three curves of each group; in each group the center line corresponds to rated maximum condensing water temperature, the upper one an extreme maximum 5 F hotter and the lower one an average 5 F lower.

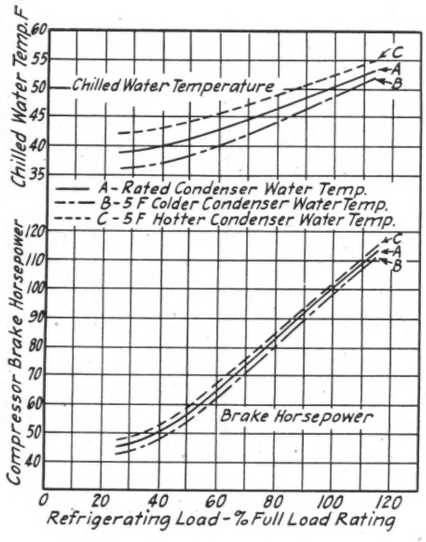


Fig. 5 — Operating characteristics of centrifugal vapor unit

Horsepower and chilled-water temperature are seen to be approximately parallel straight-line functions of the tonnage.

Constant Chilled-Water Temperature: If constant chilled- or return-water temperature is desired in a process, this can be readily obtained by mixing. For example, suppose that the full-load conditions are 500 gpm of water cooled from 55 to 50 and that 55 F return water is desired at a part load; assume further that this part load corresponds to the 45-F capacity point of the machine; then only part water is admitted to the evaporator and chilled from 55 to 45 F the remainder of the 55-F water being mixed with 45-F water under thermostatic control. To make clear what is occurring it should be remembered that the quantity of chilled water has no influence of itself on the load; the tonnage is determined by the product of quantity admitted to the evaporator and the range of cooling.

In a great many processes however, the lower water temperature is highly desirable in increasing output, or improving quality. In air conditioning the lower dew-

point attainable is nearly always desirable. In all such cases the machine is simply allowed to float on the load without any bypassing or mixing.

Reserve Capacity of Centrifugal Unit

It was shown last month in connection with the steam-jet unit that only a small increase in chilled-water temperature occurs at overloads. A similar characteristic is true of the centrifugal machine. In the first place, at constant intake volume the weight of vapor compressed will increase with temperature and density as follows:

50 F — Specific Volume = 1700 cu ft per lb
55 F — Specific Volume = 1430 cu ft per lb
Per cent weight at 55 F = 1700 ÷ 1430 = 119 per cent

Actually, however, the tendency with fixed condensing water conditions is for the volume compressed actually to increase, because the higher suction pressure tends to reduce the ratio of compression demanded. The actual increase in capacity for 5-F increased chilled-water temperature is therefore more nearly 25 per cent than 19 per cent. It follows the straight-line relationship of Fig. 5. For a moderate overload the increased chilled-water temperature is very slight—a few degrees. This is due to two fundamental differences of this type of refrigeration, as compared with machines using other refrigerants.

First, the change of density with increased temperature is more rapid with water vapor than with common refrigerants. This means that for a given increase in temperature, the weight compressed by a machine of fixed volumetric capacity increases in greatest proportion, with water vapor. In the second place, where the heat of the chilled water must be transmitted through a heat-transfer surface, any increase in load must be reflected in increased temperature difference; that is, the water must be hotter relatively to the evaporating refrigerant in order to transmit the greater amount of heat. The effect of scale or fouling, which tends to decrease the capacity at fixed chilled-water temperature and increase the steepness at which the temperature increases with load, must also be considered. Fig. 6 shows a comparison of these characteristics.

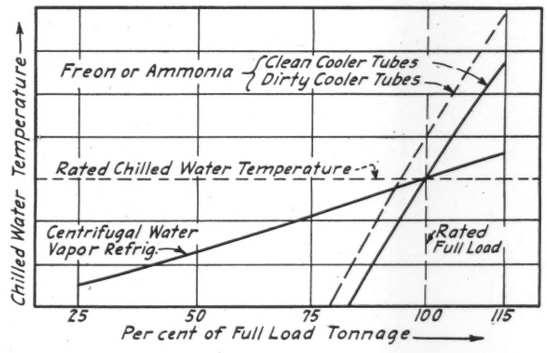


Fig. 6—Overload characteristics of centrifugal vapor unit, and ammonia or freon units

Reserve capacity against excessive temperature of condensing water with the centrifugal vapor unit has already been shown by Fig. 5 in which lines *C* are for hot water. The effect of a 5-degree increase in condensing-water temperature is a corresponding increase in condensing pressure which is reflected in higher absolute pressure in the evaporator. This increase in suction pressure reduces the vapor volume however, and the ratio of compression attainable increases; this counteracts severe increase in evaporator pressure and temperature. The chart shows the increase in chilled water temperature for 5-F excessive condensing-water temperature and it is particularly noticeable that the spread actually decreases at the heavier loads. The horsepower curves also tend to converge.

Applications of Water-Vapor Refrigeration

Standard steam-jet or centrifugal water-vapor machines are not designed for cooling brine or for producing freezing temperatures. Temperatures from 35 F up can usually be obtained but in general low temperatures should not be specified except for part load. A chilled-water temperature around 50 F is the normal rating of water-vapor units. While the overload characteristics are remarkable and highly valuable, the author wishes to emphasize strongly the futility of under-estimating operating conditions in the expectation that any machine can make up for carelessness in selection. Tonnage can be rated closely, but should not be under-estimated. Maximum condensing-water temperature should be definitely anticipated and specified; this is particularly true with cooling towers and ponds which (in the author's experience) often tend to be rated on more favorable wet-bulb temperatures than may actually prevail for appreciable periods of time. This is all the more important if the greatest tonnage is most likely to occur at times of extreme atmospheric conditions, thereby producing cumulative effects.

Industrial Processes

Highly-attenuated vapors other than water vapor can be compressed by either steam jets or centrifugal boosters; it must be remembered, however, that the discharge pressures are still relatively low, 1½ to 2 or 2½ inches, and that they are attained by condensation; it is generally necessary therefore to confine applications to condensable vapors only. It is the author's belief that other applications besides water-vapor refrigeration are possible in many industrial processes where it is desirable to raise the pressure and heat level. One such application is in low-temperature evaporation where vapor can be re-compressed and returned to the system.

Comfort Conditioning

Two large and interesting installations of motor-driven centrifugal machines will be in operation this summer, according to present plans. The one mentioned on page 164 will be installed on the 58th floor of the RCA Building in New York in connection with air conditioning four floors of office space for the American Cyanamid Company. The unit is rated at 165 tons and will operate in connection with a cooling tower.

The second installation of interest is a 190-ton unit to be installed in the Bloomingdale department store, New York City. This unit, as is the case with the RCA unit, will be driven by a squirrel-cage induction motor at constant speed.

Identify Plant Pipe Lines to Save Time, Mistakes

By **Robert W. Cuthill***

A TYPICAL piping identification chart as used at the Naval Aircraft Factory, Philadelphia, Pa., is shown here. This chart, together with drawings showing all subsurface pipe lines, has proved invaluable for quickly locating new connections and has probably prevented costly mistakes.

A chart of this kind should be accompanied with a set of written instructions for the proper painting and marking of pipes, conduits and ducts. Ours are as follows:

1. All bare piping, except brass, copper, metallic plated or galvanized, shall be painted.

2. All piping systems which are to be insulated shall be given one coat of asphalt paint before the pipe covering is applied. Superheated steam lines require one coat of special paint.

3. Galvanized ventilation ducts, exhaust vents, and dust-collecting systems shall not be painted except when required to match adjacent structure.

4. All insulation on piping inside power plants shall be painted with one coat of glue sizing and two coats of white lead and linseed oil paint. Insulation on piping outside of power plants shall be painted with two coats of asbestos paint colored to match adjacent structure.

5. All exposed risers, radiator connections, radiators, and floor and ceiling plates for heating systems shall be thoroughly cleaned and painted with two coats of paint free from linseed oil. This paint may be bronze, aluminum, or a colored enamel to match adjacent walls.

6. Smoke stacks and flues shall be painted with special black heat-resisting paint, two coats inside and one coat outside. Before painting all joints must first be sealed air-tight with flue cement.

7. Steam exhaust piping from turbines and condensers shall be painted with two coats of bitumastic enamel.

8. On all bare piping (except as noted) requiring dark colors, the first coat shall be a priming coat of red lead and linseed oil and the second coat shall be lead and linseed oil paint of the desired color.

9. All bare piping, except as noted, requiring light colors shall be painted with two coats of lead and linseed oil paint of the desired color. To insure complete coverage one coat may be slightly darker than the other.

10. All pipe lines to equipment shall be painted to five feet above floor the same color as the machines to which they are attached or the same color as adjacent walls if not connected to machinery.

11. The service of each piping system is to be identified by different colors, the proper painting of the flanges, or by stripes as indicated on the color chart.

12. All stripes shall be painted on the pipes or insulation at right angles to their center line and the edges shall be sharp and clean.

13. On pipes passing through floors identify by painting above and below each floor, except below ground floor.

14. On pipes passing through walls and partitions identify by painting on each side, except beyond outside walls.

15. On long runs of piping identify in conspicuous locations about fifty feet apart.

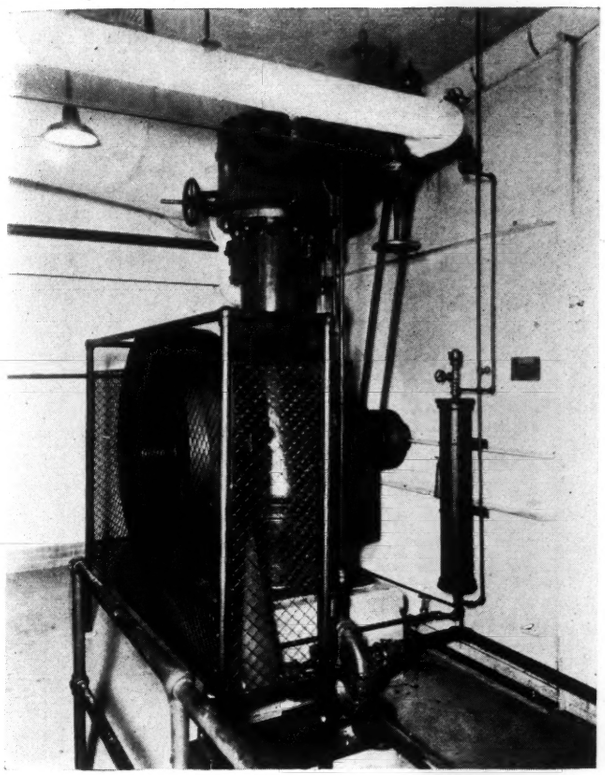
16. Identity painting may be located at other points as considered advisable for correct identification of the piping systems.

17. When pipe lines are brass, copper, metallic plated, galvanized or insulated three identity color stripes are to be painted in accordance with the color chart.

18. When it is not desired to paint a complete piping system three identity color stripes are to be painted as specified in the color chart.

STEAM PIPING	1	SUPERHEATED STEAM	White / Black / White (Flange) — 6" — 6" — White / Black / White — 6" — 6" — 6"
	2	SATURATED STEAM	White / Green / White — White / Green / White
	3	EXHAUST STEAM	White / Vermilion / White — White / Vermilion / White
RECEIVER PIPING	4	HIGH PRESSURE DRIP LINES TO HOLLY RECEIVER OR TRAPS	Orange / Black / Orange — Orange / Black / Orange
	5	HOLLY RECEIVER TO BOILER OR DISCHARGE LINE FROM TRAPS	Orange / Vermilion / Orange — Orange / Vermilion / Orange
FRESH (OR) CITY WATER PIPING	6	COLD FRESH WATER INCLUDING WATER SEALING PIPING OF TURBINES, STEP BEARING PIPES	Dark Blue / Vermilion / Dark Blue — Dk. Blue / Vermilion / Dk. Blue
	7	WARM FRESH WATER (SUPPLY & RETURN) DISCHARGE FROM HOT WELL PUMPS TO HOT WELL, HOT WELL TO FEED WATER HEATERS	Dark Blue / Black / Dark Blue — Dk. Blue / Black / Dk. Blue
BOILER PIPING	8	BOILER FEED WATER, FEED WATER HEATER TO BOILER DRUM	Pompeian Red / Pompeian Red / Pompeian Red — Pompeian Red / Pompeian Red / Pompeian Red
WASTE LINES	9	BLOW-OFF, LOW PRESSURE DRIPS, OVERBOARD DISCHARGE, OVERFLOW AND OTHER WASTE LINES	Lead Color / Black / Lead Color — Lead Color / Black / Lead Color
SALT OR RIVER WATER PIPING	10	SUCTION TO FIRE AND CONDENSER CIRCULATING PUMP	Lt. Green / Lt. Green / Lt. Green — Lt. Green / Lt. Green / Lt. Green
	11	DISCHARGE FROM CONDENSER CIRCULATING PUMPS	Lt. Green / Black / Lt. Green — Lt. Green / Black / Lt. Green
	12	DISCHARGE from CONDENSERS	Lt. Green / Vermilion / Lt. Green — Lt. Green / Vermilion / Lt. Green
BEARING OIL SUPPLY	13	SUPPLY TO TURBINES AND AIR COMPRESSORS	Tan / Black / Tan — Tan / Black / Tan
BEARING OIL RETURN	14	RETURN FROM TURBINES AND AIR COMPRESSORS	Tan / Vermilion / Tan — Tan / Vermilion / Tan
FIRE MAINS	15	DISCHARGE FROM FIRE PUMPS, SPRINKLER SYSTEMS	Vermilion / Vermilion / Vermilion — Vermilion / Vermilion / Vermilion
PNEUMATIC PIPING	16	COMPRESSED AIR PIPES	Black / Black / Black — Black / Black / Black
VACUUM PIPING	17	VACUUM	Yellow / Black / Yellow — Yellow / Black / Yellow
ELECTRIC CONDUITS	18	ELECTRIC CONDUITS	Color same as adjacent surface — Color same as adjacent surface
FUEL OIL PIPING	19	SUPPLY	Tan / Green / Tan — Tan / Green / Tan
GASOLINE PIPING	20	SUPPLY	Brown / Yellow / Brown — Brown / Yellow / Brown
	21	RETURN	Brown / Blue / Brown — Brown / Blue / Brown
DRAINAGE PIPING	22	WASTE LINES, DRAINAGE AND SEWER PIPING	Black / Pink / Black — Black / Pink / Black

*Assistant Chief Draftsman, Naval Aircraft Factory, U. S. Navy Yard, Philadelphia, Pa.



Single-cylinder, double-acting, 18x12-in. engine driving a large ventilating fan in a St. Louis school. Taking steam at 30 lb gage, the engine exhausts to the heating mains 90 per cent of the heat originally available in the steam

By **Perry West***

IN PRACTICAL MANNER, Mr. West continues in this article his explanation of the "why and how" of utilizing exhaust steam. Previously, the general considerations involved were discussed, and curves were developed for studying the steam required for space heating and the exhaust-steam supply. This month the heating requirements of buildings, and the use of exhaust steam for water heating, drying, process, and air conditioning are considered.

Exhaust Steam Utilization Offers

Space-Heating, Air-Conditioning, Drying, Water-Heating, and Process Economy to Plants and Buildings

THE general considerations involved in the utilization of exhaust steam for space heating, air-conditioning, drying, water heating, and process work were discussed by the writer in the February H. P. & A. C. The use of exhaust steam for space heating was discussed in detail, and curves for studying heating loads and the exhaust supply were developed and shown. This month, we continue our study.

Heating Requirements of Buildings

After plotting the available exhaust the next step is to determine the heating steam requirements in terms of lb of steam per hour for each of the conditions for which the available exhaust is plotted. The yearly heating requirements for different classes of buildings (in lb of steam per sq ft of direct cast-iron radiation, or its equivalent, per heating season) may be estimated from the following data for the latitude of New York City, which has 5300 degree-days per season; for other locations, values proportionate to their degree days should be used:

For manufacturing or commercial loft type buildings where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a. m. to 5 p. m. No Sunday or holiday use and no night use. Factor: 325 lb.

For office buildings using steam during day hours only to maintain 70 F from 9 a. m. to 6 p. m. for approximately 240 days (heating season). No night use. Factor: 400 lb.

For office buildings using steam during day hours and at night when required to 7, 8 and 9 p. m. (customary where there are stock brokers or banking offices), 204 days. Factor: 500 lb.

For residences of the block type (not detached) where high-class heating service is required somewhat similar to apartment buildings. Factor: 550 lb.

For apartment houses where high-class heating service is required (steam off at midnight). Factor: 650 lb.

For hotels (commercial type) where very high-class service is required; 24-hr service. Factor: 800 lb.

For schools without mechanical ventilation. Factor: 500 lb.

For schools with mechanical ventilation. Factor: 350 lb for direct radiation plus 90 lb for direct equivalent of indirect air heaters.

For university dormitory buildings. Factor, 650 without heat controls; 500 with heat controls.

For university educational buildings. Factor: 775 without heat controls; 575 with heat controls.

*Consulting Engineer, Newark, N. J.

The total heating requirements per season may be divided between the months from October to May inclusive, in accordance with the following percentages: October 6%, November 8%, December 16%, January 20%, February 20%, March 15%, April 10%, and May 5%.

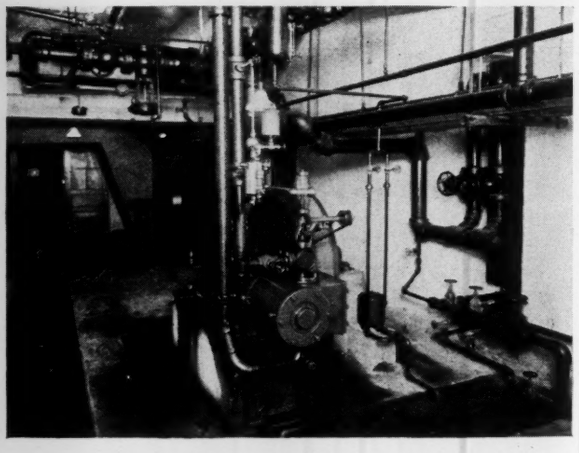
The maximum load for the four months of coldest weather may be taken as 0.3 lb of steam per sq ft of cast-iron radiation or equivalent per hr, and the corresponding loads for the four milder months may be taken as 40 per cent of that for coldest weather. The average load for the season may be taken as 40 per cent of the maximum. The daily and other variations will depend somewhat upon the use of the building, but the curves shown in the February, 1934, H. P. & A. C. may be used as a guide in the absence of more definite data.

As a matter of fact, any definite data available for making up curves for heat requirements should be used in preference to this method of estimating, but inasmuch as such definite data are usually lacking, some such approximation as this usually must be employed.

Balancing the Loads

After curves for the exhaust available and the steam required for heating are developed they can be superimposed and compared. If the exhaust available is in excess, the following methods of adjustment should be studied and the most economical adopted.

1. *Lowering the back pressure:* This will reduce the steam rate of the units and a point might be found where the loads will balance on this basis. A vacuum system of steam heating operating below atmospheric pressure may be used to advantage by carrying high vacuum when the heating loads are light, and lower vacuum—or even operating above atmospheric pressure—when the heating loads are heavy.



Engine-driven ventilating blower unit in Morgan Park High School, Chicago. There are 669 engine-driven blowers in use in the Chicago public schools, 334 of which are belt driven and 335 direct-connected

2. *Condensing and bleeding:* By operating a turbine condensing and bleeding steam out for light heating-load periods and operating the turbine non-condensing and using exhaust for heavy heating-load periods. Or operating the turbine or engine condensing under low vacuum and taking a portion of the exhaust for heating under the same vacuum.

3. *Storing of heat:* By means of hot water accumulators for use with mechanically-circulated hot-water heating systems to flatten out the peaks in the exhaust supply.

4. *Use of condensing exhaust turbines* for utilizing the surplus exhaust steam with a pilot-operated balanced back-pressure valve between the prime-mover exhaust and the low-pressure turbine so that any predetermined pressure can be maintained in the line from which the steam for heating is taken.

If the exhaust supply is deficient the following methods of adjustment should be studied:

1. *Raising the back pressure* so as to increase the steam rate of the units.

2. *Bleeder turbine* from which the exhaust is used together with such a quantity of steam bled from a point as near the exhaust end of the turbine as will produce the total quantity required for heating.

3. *Supplementing with live steam.* This involves the use of the exhaust as far as available and the supplementing of this with live steam through an automatic pressure-reducing valve.

For studying these various methods certain data regarding the water rates of units under bleeder and condensing conditions must be obtained from the manufacturers for each particular case.

It may be said in general that substantially as much power may be derived from steam from atmospheric pressure down to 29 in. vacuum as may be derived from the same steam from 100 lb down to atmospheric pressure, and the ratios are somewhat in proportion to these pressure ranges.

Water-Heating Loads

In many buildings a considerable quantity of exhaust steam may be used for heating water. A general idea of the magnitude of this load may be had from the following figures showing the steam required for heating water per year in terms of the percentage of steam required for heating the building per season.

Office buildings	10 to 25%
Hotels	50 to 70%
Apartment houses	25 to 75%
Hospitals	25 to 60%

It should be borne in mind that these percentages are for 12 months water heating as compared with from 6 to 8 months heating loads. Daily variations in hot water loads are illustrated in the curves reproduced with the February article.

One pound of low-pressure steam is generally allowed per gallon of water heated from 60 to 175 F; hot-water loads are plotted and either added to the heating load or treated separately.

In hospitals and hotels the refrigerating and hot-water loads substantially balance as far as exhaust available from steam-driven refrigerating machinery and steam required to heat water is concerned. For this reason it is frequently advisable to use steam-driven refrigerating machinery and use the exhaust for heating water, although the remaining power and light may be purchased.

Any steam to be used for feedwater heating must also be added to the heating load or be deducted from the exhaust so as to show that available for other purposes.

Use of Exhaust Steam for Drying and Process Work

Exhaust steam may be used in most drying processes although live steam is frequently used for the reason

that less heating surface and a smaller volume of air circulation is required. By properly designing the heating surface and the air movement with the line of passage of the material, most drying processes can be designed or remodeled to operate economically on exhaust steam if this can be made available.

Drying or other process work actually requiring temperatures above those of normal exhaust steam may be taken care of by increasing back pressures, or in cases where the required increase in back pressure would produce excessive quantities of exhaust, the initial steam may have its pressure raised or be superheated. The loads may be balanced somewhat the same way as for heating with the exception that the increasing of initial pressures and superheating of the steam fed to the prime movers will usually be found more applicable.

Using Exhaust Steam for Air Conditioning

Another service which offers an opportunity for using exhaust steam is air conditioning. The necessary heat for warming and humidification in winter and for reheating in summer can be supplied directly from exhaust steam. Refrigerating effect for cooling and dehumidification may be produced by the ammonia absorption process, or by the steam-jet refrigerating process, using exhaust steam in either case.

Ammonia absorption processes require about 30 lb of exhaust steam per hour per ton. The steam-jet machines may require about 60 lb of exhaust steam per hour per ton. One ton of refrigeration is the absorption of 200 Btu per minute.

Broadly speaking, the heat requirements for air conditioning consist of 1 Btu for every 55 cu ft of air per degree rise in temperature plus 970 Btu for every pound of water evaporated in the process of humidification. For cooling and dehumidifying the Btu requirements are the same.

For ordinary winter weather in our northern climate the outside air may be taken at an average of 30 F and will require the addition of about 3.5 grains of moisture per cu ft. (There are 7000 grains to the avoirdupois pound).

The quantity of air to be taken in from the outside is 7.5 to 10 cfm per occupant or from ¼ to ⅓ of the total air circulated on a basis of 30 cfm per person or not less than one air change of total air circulated every 10 minutes.

Ordinary summer conditions are 95 F in northern climates and 100 F in southern climates with 3 to 4 grains of moisture to be removed per cu ft of air from the outside; inside temperatures to be from 5 to 8 F lower than outside.

In estimating air conditioning loads for either winter or summer conditions the heat and moisture given up by the occupants, the heat given up by lights, motors, etc., the heat loss from the building in winter and the heat intake of the building in summer, must be taken into consideration.

The heat from occupants is 275 Btu sensible heat and 0.1 lb of moisture corresponding to 100 Btu additional, or latent heat, per occupant per hour.

The heat from lights is 3415 Btu per 1000 watt-hours and the same from motors per hp-hr.

The heat input to buildings is about 3 Btu per hr in northern climates and 5 Btu in southern climates per sq ft of ordinary wall; about 150 to 200 Btu per hr per sq ft of window without awnings and from 30 to 50 Btu with awnings from sunshine and about 15 to 20 for windows without sunshine. Roofs are a variable quantity, but may be figured at from 10 to 20 Btu per sq ft per hr.

The figures for cooling requirements are for maximum loads and average loads may be taken at 50 per cent of maximums. The seasons of air conditioning range from 4 months in the south to 8 months in the north for heating and from 6 months in the south to 3 months in the north for cooling.

In considering the use of exhaust steam for refrigeration it must be borne in mind that the greater requirements for water with these systems as compared with compression systems often make the cost of operation prohibitive.

Perhaps one of the most economical arrangements for the use of steam in air conditioning is to use turbines for driving the fans and/or refrigerating machine, the exhaust from the turbines to be used for heating and humidifying the air in winter and for reheating the air in summer.

FLOW AND PRESSURE-DROP Calculations for Process Piping

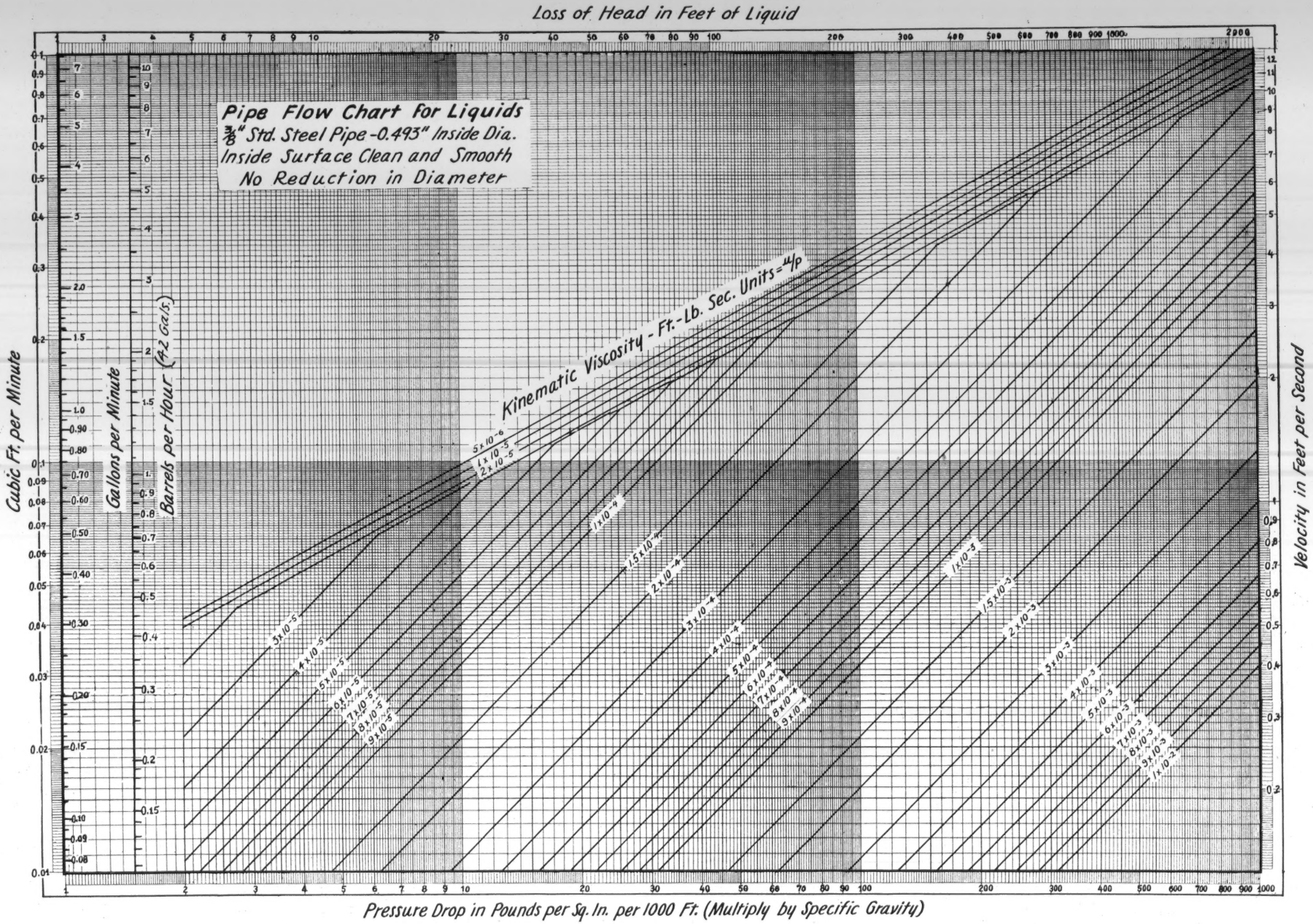
By **Emory Kemler***

ON the next page is a chart designed to facilitate the making of pressure-drop calculations for liquids of various kinematic viscosities flowing in ⅜-in. pipe. The complete series is planned to cover standard pipe sizes from ¼ to 4 in., the viscosities being chosen to cover a range that will take care of any ordinary process fluid that is being piped.

The first two charts to be published appeared in the October, 1933, issue and were for 1-in. and 2-in. sizes. A chart for 1¼-in. appeared in November, 1½-in., December, ¾-in., January, ½-in. March. An explanation of the charts and an example of their use also was published in October, as was a conversion chart for kinematic viscosity and viscosity in Saybolt seconds. Allowing for roughening and accumulations and turbulent flow were also discussed.

This explanatory material in the October issue should be referred to in connection with use of the chart published this month.

*University of Pittsburgh, Pittsburgh, Pa.


Loss of Head in Feet of Liquid
Pipe Flow Chart for Liquids
3/8" Std. Steel Pipe -0.493" Inside Dia.
Inside Surface Clean and Smooth
No Reduction in Diameter
Kinematic Viscosity - Ft.-Lb. Sec. Units = μ/ρ
Cubic Ft. per Minute
Gallons per Minute
Barrels per Hour (42 Gals.)
Velocity in Feet per Second
Pressure Drop in Pounds per Sq. In. per 1000 Ft. (Multiply by Specific Gravity)

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

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Air Conditioning in its Relation to Human Welfare

By C. A. Mills,* M. D., *(NON-MEMBER)*, Cincinnati, O.

AS this session of our 40th Annual Meeting is devoted to the health aspects of work in heating and ventilating engineering, I feel hesitant in expressing some ideas of mine, but am led to accept the invitation to do so because of my great interest in the problems involved. For the past six years I have been studying the relation of our climatic and weather environment to our health and general welfare, and the results of these studies have convinced me that here is a factor of very great importance in our lives—one, too, which has been badly neglected in the scientific advance of the past 50 years along public health lines.

Engineers have really accomplished wonders in the utilization of inanimate force, making almost anything possible by the mere turn of a switch. You have brought our indoor environment almost completely under control, except for changes in barometric pressure and ionization. Even these will, I believe, soon be conquered by your ingenuity. But how about the other side of the picture—with everything physical within our grasp, do we know just what we want, or what is best for us? Unfortunately medical and physiological research has sadly neglected this field in past decades. There is, however, now taking place a quickening of interest along this line, together with a realization of the tremendously important place our climatic environment has in our existence.

Let me briefly summarize for you a few of the known facts about man's dependence on his atmospheric environment. Under carefully controlled laboratory conditions, it has been found that animals adapted for a few weeks to constant moist heat lose in large measure their heat producing capacity and bodily vigor. When subjected to chilling emergencies, they cannot quickly increase their heat production, and so suffer a fall in body temperature and are prostrated. Hand in hand with this sluggish heat and energy metabolism, they show a lowered resistance to infection. Pneumonia, tuberculosis, and other types of infection attack them with greater ease than is seen in more vigorous animals. Under the greater stimulation of a cold environment, animals are more lusty, lay on more fat, are better able to resist sudden chilling and are more resistant to all forms of infections. Most active of all are animals shifted daily from heat to cold, so that they are required to make frequent and sudden adjustments in their heat production. Such animals stand chilling best of all, but are most sensitive to high temperatures.

These differences induced in laboratory animals by artificial climates exactly parallel differences observed in man under varying climatic conditions. In our cool stormy regions of the Temperate Zone man lives on a high energy plane. He is vigorous, full of pep and vitality, and must be always doing something. In the Tropics and Orient, and in our own Gulf states, there is much less energy available, so that a much greater part of the daily supply must go into the business of mere existence, leaving little for racial or economic advancement. Southerners coming north show their sluggish heat metabolism by the way they chill under conditions that natives call comfortable. For indoor comfort they demand a temperature 4-8 F higher than we need.

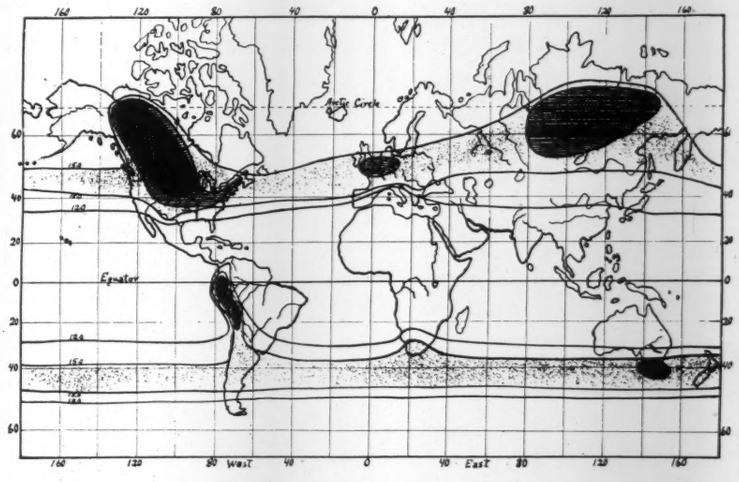

Fig. 1—World Map Showing Intensity of Climatic Drive

I have constructed a world map based on the intensity of climatic drive imposed on man in the different regions. This is not the place to present the details of calculations used in deriving the indices of climatic stimulation. Suffice it to say that the map, Fig. 1, is based on the best available knowledge we have as to the effects of this climatic drive on man and animals, and represents in a fairly accurate way comparative differences over the earth. It will be seen that we here in north central North America have imposed upon us the most vigorous climatic drive man is anywhere called upon to endure. Perhaps this in a measure explains the astonishing rate of development of the physical resources of the continent. Perhaps it also explains our restless and impetuous zeal for action, as well as our irritated, nervous state. In our Gulf States the climatic drive is very much less, similar to that of the Mediterranean coun-

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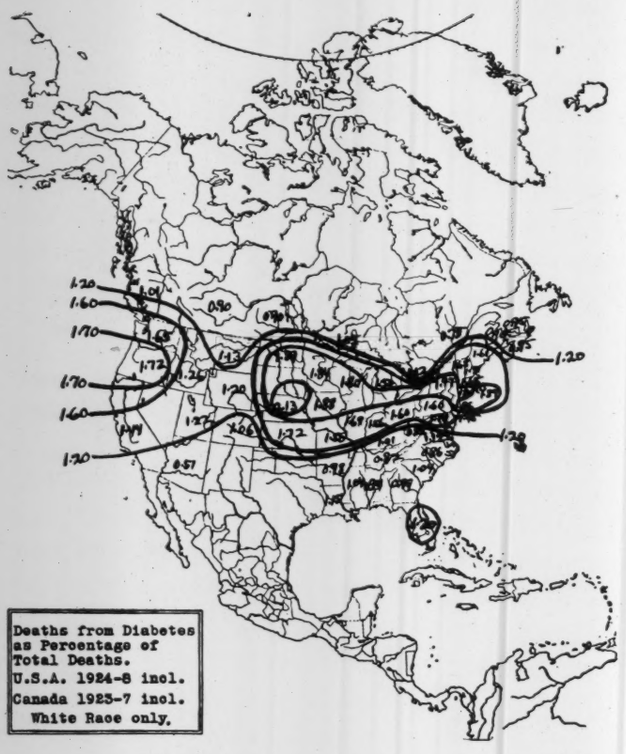

Fig. 2—Deaths from Diabetes as Percentage of Total Deaths
U. S. A. 1924-8 incl.—Canada 1923-7 incl.—White Race only

tries of Europe and of Japan and North China.

We here in the North have one rather serious situation to face in regard to this climatic vigor that gives us our abundant energy. Unfortunately we are not capable of unlimited response to this stimulation, and are showing definite signs of breaking under the strain. Certain of our glands of internal secretion are closely bound up in our energy metabolism and when these fail, disease appears. We derive most of our heat and energy from the burning of glucose, and in this burning the pancreas plays an essential and important part. Thus diabetes is only a sign of pancreatic failure to meet the level of activity demanded. Toxic goiter also represents failure of successful adaptation to stimulation, and we might well class arteriosclerosis and high blood pressure in the same category. The accompanying maps, Figs. 2-5, show how closely these bodily disturbances follow the climatic drive. Diabetes in the North is a very much more severe disease than it is in the South, reaching its greatest severity in our west central states where the climatic stimulation is most intense. The same holds true for exophthalmic goiter, for pernicious anemia, for arteriosclerosis and a number of lesser metabolic diseases, and curiously enough for cancer.

These advancing signs of bodily breakdown in the most stimulating areas, are more than matched by the mental picture. Suicides, mental breakdown, and nervous disorders are nowhere so frequent as here and in the most stimulating European regions. These bodily and mental diseases mentioned here are the ones causing greatest concern to medical and public health authorities of today. It is essential, though, that they recognize the basic role played by climate if there is ever to be intelligent handling of the situation.

We see, then, that the South profits by its lessened climatic drive in maintaining greater bodily and mental stability. The easier, more relaxed and care-free existance that goes with the lower energy level less often brings on the diseases of exhaustion. In many ways the southerner is to be envied. With his lower energy state, however, he is more susceptible to infections and shows a higher death rate from tuberculosis, acute nephritis and acute appendicitis. The accompanying map of acute nephritis (Fig. 5) death rates in North America shows just the reverse of the diabetes findings (Fig. 2). Where diabetes is most severe, there acute nephritis causes fewest deaths, and in the South, where diabetes is mild, acute nephritis is most severe. These same findings hold true for the countries of Europe also.

In addition to these disease and energy differences that come from the varying intensity of climatic drive, we have marked effects on health in our stormy regions that come as a result of the wide frequent swings in temperature and barometric pressure. Acute appendicitis attacks and suicides, for instance, have been found to come mainly with the periods of rapidly rising temperature and falling pressure that presage our storms. Colds and pneumonia, on the other hand, come with the days of rapidly falling temperature. But in all these cases, the frequency of attacks are determined by the frequency and severity of the storm changes in the weather, week after week. The central portion of North America is certainly not to be envied in these particular health aspects, and they should be considered in calculating the price of our exuberant energy.

Realization of the many important phases of human life that are affected by climate and weather changes

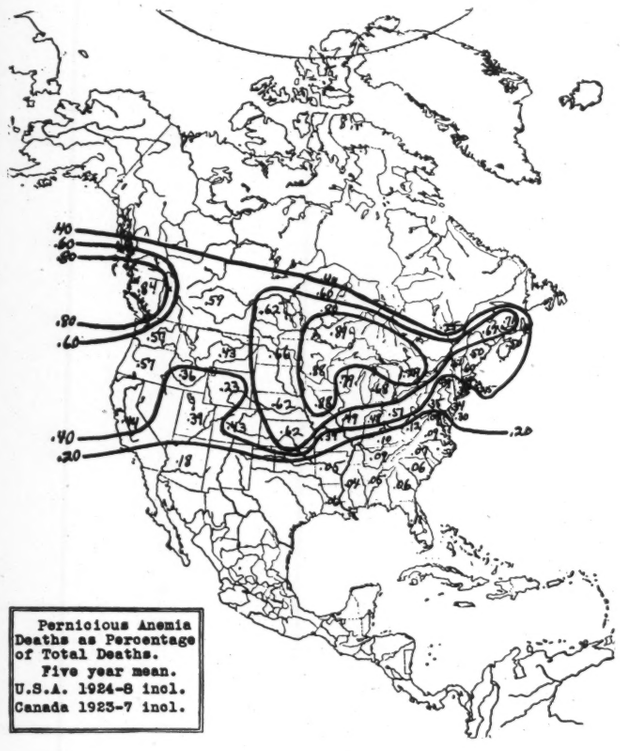

Fig. 3—Pernicious Anemia Deaths as Percentage of Total Deaths
Five year mean—U. S. A. 1924-8 incl.—Canada 1923-7 incl.

naturally brings up the question,—what are we to do about it? If, in the very stimulating areas of the earth, as in our middle western states, the human machine shows evidences of breaking under the strain, shall we not seek means of lessening the driving force to within safe limits? And in the tropics, where man is held down to a mere existence energy level by the climatic stagnation, should we not attempt to relieve him from the pall of physical depression that is holding him down? If there be a destiny for mankind to fulfil here on earth, then release from the devitalizing effects of tropical and oriental moist heat would enable those hundreds of millions of people to attain that destiny more quickly and certainly. Should not the people of our own Gulf States be given the chance at an energy level that would permit them a more fair competition with the energetic northerners?

But when we ask ourselves these questions, we should also keep in mind certain others that come as natural corollaries. Recognizing excessive climatic drive as the principal basis of our American restlessness and inability to relax, are we justified in foisting upon less energetic peoples this urge to action which will largely destroy their present calm and carefree existence? For there is little doubt but that raising the physical energy level entails always a more irritable nervous system, with less complacency and contentment. It does seem, however, that the very energetic level of life here in America makes us assume that such a state would be best for the whole world. Hence out go our commercial and educational, as well as religious, missionaries to all the backward corners of the earth. The people of China, Patagonia and Greenland shall not live on without modern plumbing and automobiles if we can help it.

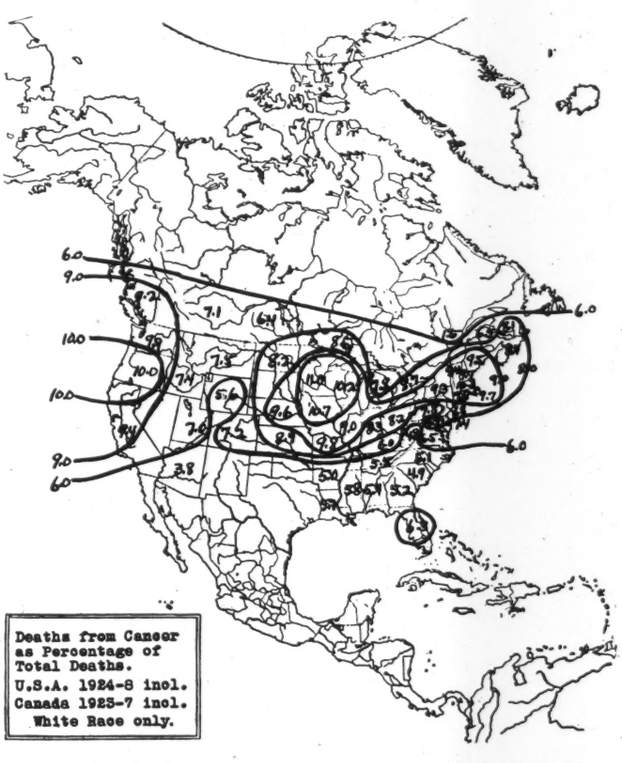

Fig. 4—Deaths from Cancer as Percentage of Total Deaths
U. S. A. 1924-8 incl.—Canada 1923-7 incl.—White Race only

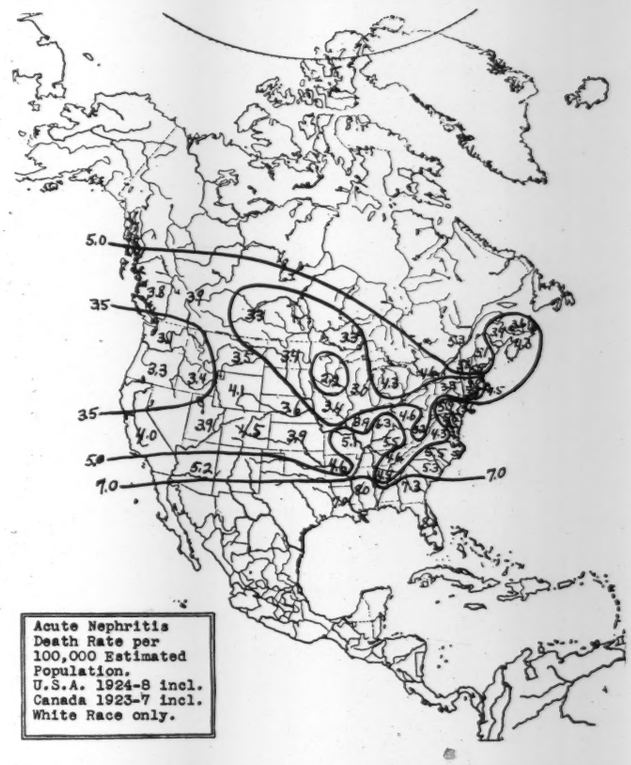

Fig. 5—Acute Nephritis Death Rate per 100,000 Estimated Population
U. S. A. 1924-8 incl.—Canada 1923-7 incl.—White Race only

Probably most important of all for us here in America is the human stress that comes from our severe and frequent storm changes. The wide swings in temperature and barometric pressure which we must endure every few days do us untold harm. True enough it is these changes that generate in us our high level of energy, but they also bring us to grief and bodily misery. Such diseases as pneumonia, acute appendicitis, "colds," sinusitis, and chronic arthritis—these all are closely related to our severe storm changes and do much to reduce our health and efficiency. Effects on the nervous system are perhaps even more important, for we must be constantly shifting from the irritation and feeling of futility that comes with the low pressure-rising temperature days, over to the exuberant exhilaration of the cool high pressure periods. We are never left in a state of calm for more than a few days, but must be always jumping about mentally, due to the effect upon us of these weather changes. It is no wonder we are notional and take up every fad that comes along. Sub-conscious realization of this irritated mental state leads us into the passage of innumerable laws and regulations, most of which we disregard as soon as they are passed. It was felt, for instance, that legal prohibition of alcoholic beverages would do away with most of our wild life in society, but quite the opposite occurred; and now, with the return of beer, a feeling of contentment seems to have settled over the nation.

One cannot stress too much the deleterious mental effects of our frequent weather changes. Continual confusion and changing from depression to exhilaration exhausts us emotionally and leads to an existence full of irritation and restlessness. That is why people from

the stormy area find Hawaii, Southern California and Florida such desirable places for rest and relaxation. Particularly do we need such climates as these during our late winter and early spring months, when the northern storminess is greatest and many of us are near physical exhaustion. In our recent years of prosperity it began to look as though there would develop a migratory movement in the population somewhat similar to the seasonal flight of many birds, and perhaps such seasonal migration will make further headway in years to come. There will, however, always be many millions who cannot get away from home and must remain blindly submissive to the vagaries of the weather unless artificial control is made available for them.

Our greatest hope of climatic control, outside of seasonal migration, lies in the new and rapidly developing industry, air conditioning for human comfort. Through it, and the newer methods of construction that give almost complete insulation from the outdoor environment, we are now able to exercise almost complete control over temperature, humidity and suspended dirt particles in our homes, offices, factories, and public buildings. The cost of the necessary equipment for installation in old buildings is still beyond the reach of many, but installation in properly planned new structures is in the long run more economical than our old haphazard methods of heating and ventilating. It has been shown, for instance, by hospital architects that hospitals, fully adequate for every reasonable need, can be built and equipped for complete temperature control the year round for not over $2,500 per bed, whereas past costs have ranged from $4,000 to $10,000 per bed, with no provision for cleaning the air or for summer cooling.

It has been pointed out that not only do the heat waves of summer bring on a variety of acute ailments, such as acute appendicitis and summer diarrhea, but that these patients do most poorly when compelled to lie in hot hospital rooms. Surgeons are aware of this lessened vitality of patients during the periods of heat and tend to postpone until cooler weather major operations that are not of an urgent nature. Most often, however, delay is not possible, as in acute appendicitis, and on this account it would seem imperative that surgical hospitals should be fully protected against excessive heat, for both the operation and the after care. It is quite likely that convalescence from operations and fever ailments could be considerably shortened by providing the proper mean temperature level and proper daily range for each day to give the right degree of stimulation. At present, throughout the Tropics and Orient, recovery from a serious illness or operation is much slower than in more stimulating regions. Let us hope that it is only a matter of time until hospitals and the medical profession will be brought to greater use of air conditioning methods as direct aids in securing recovery from illness.

In lieu of expensive equipment in homes, or of seasonal migration to counteract climatic effects, it would seem advisable that there be made available in each city artificial climates of various types, in hotels, hospitals or sanataria, where the patients can go and enjoy the relaxing effects of Florida or Hawaii for a couple of weeks and still be near business or professional interests. Two weeks' rest in bed, in a room kept at 85 F, with 70 per cent relative humidity, would go a long ways toward quieting down the over-dynamic hypertensive individual of our northern cities, and no doubt great use would be made of such facilities when people came to a clearer realization of whither they are heading in our area of greatest storminess and most intensive climatic drive. The same type of air conditioning equipment could be used, also, to produce changeable stimulating environments of any intensity desired in speeding recovery from run-down states, or for use in relieving hay fever and asthma sufferers during their periods of distress. Tuberculosis hospitals could, in any location, be given the same advantageous climatic effects as are offered by the best mountain sanatoria. In fact we can have only a faint realization at present as to the great possibilities such artificial climates may some day offer in promoting human health and function. The field is yet too new and untried to allow of a close evaluation of its possibilities. Its potentialities, however, are alluring.

Another great future possibility is the use of cooling devices in the warmer countries to step up human energy and efficiency to a level more nearly equal to that of the cooler temperate regions. Animal experiments indicate that this can be readily accomplished by just a few hours of cooling each day, such as might be most economically obtained by chilling the sleeping quarters. Everyone is aware of the refreshing effect of a cool night after a hot day, and how the day's heat does not depress nearly so much when a good night's sleep is made possible. It would be a most interesting experiment in mass effects to try such night cooling of a sample population for a few summers to see if their bodily energy and vigor responded as did that of the laboratory animals. Research engineers are even now working to perfect a less expensive type of cooling device that might bring such an experiment within the means of the people. It must be remembered, however, that the blessings of this higher energy level are not entirely unmitigated. The increased efficiency and vigor carry with them a more restless and irritative existence—a loss of the southern complacency that makes life so pleasant and carefree. Also such a change would render the population more sensitive to the summer moist heat, as well as probably increasing their tendency to diabetes, arteriosclerosis and the like. The American of today, however, usually disregards these dangers, which to him are vague and probably aimed at somebody else anyhow, and grasps at any thing that promises greater wealth. So it is likely that eventual use will be made of the possibilities here outlined for increasing the productiveness of man in the less stimulating climates so that he may compete on more even terms with his more energetic brothers.

Another important aspect of the use of artificial climates is that concerning the breeding and care of domestic animals. Experimental studies have shown that animals raised under optimal climatic conditions are more fertile, mature earlier and are in all ways more vigorous than are those subjected to moist heat. With laboratory mice the differences in fertility and growth are marked. Litters are usually almost twice as large in the cold as in the hot environment, and in addition they are much more lusty. With the small litters of the hot room, a large number are born dead or die soon after birth. In addition, although the animals mate freely in either heat or cold, conception is almost invariable in the cold but

difficult to attain in the heat. These findings, if applied to stock breeding would mean a very great handicap for the regions that have prolonged periods of moist heat such as is met with in the Gulf states.

Just as important as the fertility differences, however, are the changes in rate and type of growth produced by the different environments. By proper stimulation, growth is at all times far ahead of that seen under moist heat. The animals raised in the cold are more plump, their bodies are larger, and their weight markedly higher. The accompanying photograph, Fig. 6, of representative mice from the two groups brings out this

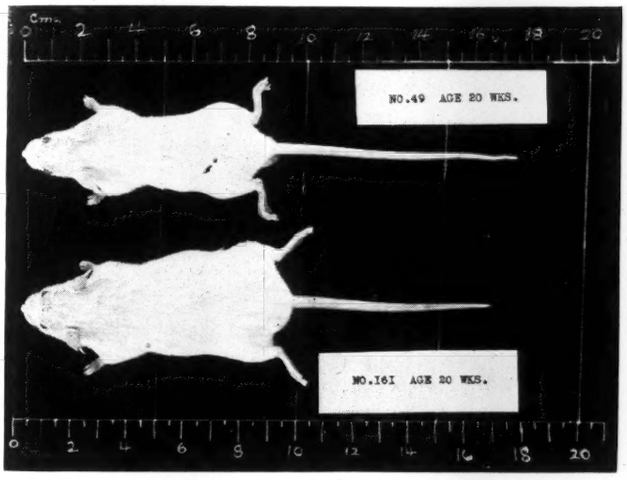

Fig. 6—Development of Mice in Hot and Cold Environment at Age 20 Weeks

point very well. The total body length, including the tail, is always greater in the hot environment, but the marked difference in body size is striking. These facts should be of great value in the raising of stock for meat, whether it be beef, pork, mutton or fowl.

The few fortunate areas of the earth where meat supplies are most easily and cheaply produced are also those characterized by a stimulating climate. The States of our North and Western Plains, Australia, the Argentine, Central Europe, the Mexican plateau—these all are favored by a climate that induces great vigor of body, rapid growth and great fertility.

The Tropics and Orient, on the other hand, and also the states of the Old South (from Louisiana eastward) are held down by devitalizing moist heat for all, or a large part, of each year, and so find domestic stock raising difficult. Breeders have tried, time after time, the introduction of the fine vigorous strains from the more stimulating regions, only to find them soon sinking to the lean stringy type so common in those regions. It is almost impossible to keep up a herd of good milk cows there, principally because of the toll taken by tuberculosis and the climatic suppression of metabolism.

Is it not time that the knowledge now available be used to liberate the South (and all regions afflicted with moist heat) from this serious handicap to their economic development? Why should they not provide their own supply of good meat, eggs and milk in abundance? Artificial cooling, properly applied, would I am sure solve the problem for them. Dog raising, poultry growing, and the proper care of race horses demands just as much consideration of these climatic effects and the means of their relief. These few remarks will serve to give you some insight into the possible future expansion of the air conditioning industry for both man and animal shelter.

Whether we will eventually be able to escape the disturbing effects of our wide swings of temperature and barometric pressure as the major storms pass by is questionable. The more time we spend each day in an environment artificially regulated, the nearer could we come to this goal. But with our goings and comings, together with our modern ideas on fresh air while we sleep, we are exposed to the uncontrolled vagaries of the weather for at least half of winter, and to far more than half during the rest of the year. Are we headed for a more rat-like burrowing existence indoors as the decades pass, or by natural selection are we to permit the climatic drive to eliminate those who cannot adapt to its severity? The latter is a matter of centuries and beyond our control, while the former is even now in actual process of accomplishment, with rapid strides being made in achieving independence of the external weather changes.

The future likely holds for us some striking changes in our idea as to the value of fresh air. I have little doubt that another decade will see great care being exercised to avoid excessive climatic stimulation during our cooler seasons, and one of the first fads to go will be the sleeping with wide open windows. The day is surely coming when we will intelligently seek to smooth off the severe weather changes here in the North, and to raise the energy level in the South by night time cooling through the warm season. Our aim should be to relieve man everywhere of the handicaps imposed by climate, by securing a universal mean best suited to our bodily capacity.

Chemical Engineers' Handbook

The new Chemical Engineers' Handbook, published by the McGraw-Hill Book Co., New York, N. Y., and prepared by a staff of specialists of which John H. Perry, Ph.D., was editor-in-chief, exemplifies in its 2,600 pages the vast amount of general engineering information which enters into the pursuit of any of the engineering professions in this age.

To the heating and ventilating engineer the Chemical Engineers' Handbook is useful because, with the development of refrigeration as an integral part of heating, ventilating and air conditioning, the various chemical reactions and the characteristics of refrigerants, absorbers and adsorbers, as well as the heat balances set up by convection, conduction and chemical reactions, are essential to a thorough knowledge of the basic methods of procedure.

The edition contains a number of interesting sections not common to most handbooks of this character, covering reports and report writing, bibliography, and a new section on patents and patent law.

This volume adequately supplements the Mechanical Engineers' Handbook issued by the same publisher.

Heating Buildings With Hot Water

By **Benj. F. Burt*** ***(NON-MEMBER)***, **Rochester, N. Y., and**

Samuel R. Lewis** ***(MEMBER)*** **Chicago, Ill.**

IN the days when many good residences were heated by hot water under gravity, or thermal circulation, the systems gave excellent satisfaction, circulation to each radiator usually being even and the heat delivered to the rooms being proportional to the intensity of the fire in the boiler. Occasionally buildings are encountered today, where the hot water heating system is upward of 50 years old, and is functioning as well as ever, with little or no repair expense during the years of service.

As civilization advanced there came a demand for a less expensive and less artful type of heating and the use of hot water circulation was neglected. Single-pipe or air vent low pressure steam heating became popular. It was more easily installed than thermally circulated hot water heating and as the boiling point of water was the lower temperature limit of the steam system, radiators smaller than those needed with hot water could be used. In the early systems the radiators were filled with steam in mild weather, and rooms were overheated; consequently windows were opened and heat was wasted. No graduated regulation of temperature by valves was possible.

People therefore hailed the advent of the patented vacuum method of double pipe steam heating. It did away with the hissing air valves of the single pipe systems. One could, if the radiators had both top and bottom connections between the sections, regulate the room temperature by valves. If the piping and the valve packing were tight the temperature of the evaporated water inside the pipes and radiators could be reduced from the 215 F of the single pipe steam system to perhaps 160 F by maintaining a pressure below atmospheric. This pressure and temperature, if in effect on the water surrounding the fire in the boiler, would promote fuel economy, since the cooler the heat absorbing material the cooler the flue gases passing up the chimney.

The saving in fuel made possible by reduced temperature of the heating medium in high vacuum heating systems has been demonstrated thoroughly. It may be accounted for to some extent by the fact that there is less opening of windows in order to cool off. The difficulty and cost of keeping these high vacuum systems in perfect physical condition year after year to maintain the high vacuum is also very real.

Vacuum systems of steam heating using pumps, and their cousins, the vapor systems without pumps, were promoted widely and the patented steam specialties used in these systems were extensively used.

In many cases with steam heating, complicated and elaborate piping systems and special building construction were necessary to protect the return pipes.

People still were neglecting hot water heating, which could provide a wide water temperature range and permit graduated room temperature control. Hot water heating may have a temperature range from room temperature to 250 F or so, and needs only two main pipes. When a circulating pump is used, all radiators, whether below or above the mains, may be served equally well.

During these many years the idea which would bring hot water heating back to serve civilization was alive but decidedly dormant. A few people remembered it and cherished it. Hydraulic forced circulation is, of course, nature's method of heating humans and animals. Thermal or gravity circulation of the blood if ever tried was abandoned by nature. Possibly thermal circulation is approached by the sluggish type of reptile during the hibernation period. The hands and feet of these reptiles get exceedingly cold during their quiescent times when the pumps are slowed down.

In man the heart is the pump for his hydraulic heating system and gives positive circulation so that if all is well the right hand is as warm as the left hand. So in hot water heating with mechanical circulation the radiator most remote from the boiler easily can be compelled to maintain practically the same mean effective temperature as that of the nearest radiator.

Nature couldn't use a rotating pump, but apparently was forced to limit her service to a reciprocating design. Man never has been able to design a reciprocating pump which approached the probable efficiency of the heart pump designed by nature, but man has the wheel and its derivative, the rotary or centrifugal pump. Man plus the lubricated wheel and its family comes very near to surpassing nature in mechanical efficiency.

With a power driven fan applied to an air stream, man can blow air far less wastefully than he can blow it by using a fire at the bottom of a chimney. With a power driven pump applied to a water stream man can circulate water far more efficiently than he can circulate it by applying heat to part of it to reduce its weight per unit of volume, so that the cooler water will fall down below the warmer water and thus start circulation.

The heat demand for every building changes constantly. At noon it may be rather light, due to occupancy by people, to work, or to the sun. At evening the heat demand may increase because of the slowing down of activities and absence of sunshine. At night the demand may decrease or cease for a time due to a lower temperature requirement. In the morning the heat demand may be very heavy to regain the night loss, since the desire will be for a higher temperature than during the night, and the sun has not yet had time for action. All of these changes may be upset by storm or by sudden changes in the heat production from people, lights or machines. In addition to daily, or rather local, changes

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Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

in temperature, the heating system must also meet the general seasonal changes. The heating season in most localities begins with an occasional slight demand for heat in September and ends with a similar demand in May. There is the heavy and constant requirement of heat in January and the varying and often widely fluctuating needs of the intermediate months. Thus to be comfortable and economical any heating system must be adjusting itself constantly to the conditions outside and inside, and if it can anticipate these conditions it will be a still better heating plant.

The smaller thermally circulated hot water heating systems accomplish this rapid adjustment. The very large thermal water heating systems are often sluggish and fairly subject to criticism. The single pipe steam systems and the vacuum steam systems gave potential quick response to the fire intensity. Any heating system which overshoots the mark will cause windows to go up to prevent overheating and will be wasteful of fuel.

It must be remembered that each room must have enough radiation to heat the space in the coldest weather. If steam is used a temperature of around 212 F will be necessary, and each radiator usually will be hot all over if hot at all. With controlled high vacuum this temperature may be reduced considerably. In mild weather with steam heating there is an almost irresistible tendency to overheat in flashes and to waste fuel by opening windows.

The response of mechanically circulated hot water heating systems to the change in heat demand by the room is as quick as the response with steam, but the controlled water temperature on a mild day may be as cool as 100 F or 120 F and the room temperature will not over-run, the windows will not be opened to relieve overheating, and much fuel will be saved.

Where old thermally circulated hot water heating systems have had faults due to inferior design or workmanship, a mechanical circulator easily can be added, with the almost invariable result that the sluggish circuits or radiators are corrected, and with the equally interesting results that fuel savings are registered.

With mechanical water circulation the water temperature difference between radiator inlet and outlet may be varied within a very wide range and the circulation rate is independent of water temperature. With thermal circulation, to produce flow there must be a considerable temperature difference between flow and return water. If there is one unfavorable circuit or radiator, all the other circuits or radiators must pass an unnecessary volume of water or must be throttled so as to carry a greater temperature difference between inlet and outlet than the sluggish circuit, so as to force some action out of the latter.

Consider water at 180 deg *flow* temperature circulating to a radiator in a room during a period of maximum demand, and that this maximum demand rate is, for this room, 4500 Btu per hour. Assume that the radiator in this room, receiving water at 180 F, will give off the necessary heat (approximately) at a mean effective temperature of 170 F, and the water will leave the radiator at 160 F. There will be a temperature drop of 20 deg; $\frac{4500}{20} = 225$ lb of water to be circulated each hour (7.5 lb per 150 Btu h). This gives no consideration to the amount of radiator surface, which of course must be adequate, and makes no allowance for losses from the piping.

A drop in water temperature of 20 deg is the customary design allowance from the boiler outlet back to the boiler inlet, but in practice some thermally circulated radiators may have a temperature drop of only 1 or 2 deg while others have such resistance to flow as to require a temperature drop of 30 deg or more to develop the necessary power to move sufficient water through them to deliver the desired heat output in Btu per hour, even at this greater temperature drop. The boiler, which must develop this power, often is handicapped by the too-rapid thermally induced flow through the more favorable circuits. With milder weather and need for water at a reduced temperature in systems like this, the unfavorable circuits consequently become more and more sluggish as the available temperature head decreases.

Mechanical circulation of the water corrects these difficulties and permits or compels a balanced circulation, with a maximum and assured temperature drop in every radiator in cold weather so that each one shall earn some value on its investment. The process of increasing the water volume or pumpage per unit of radiation, thereby decreasing the temperature difference between the flow water and the return water can, of course, be carried too far.

A zone must be established wherein pumpage is economical and where pipe sizes will be reduced intelligently to correspond, and where the temperature drop through the boiler and radiators gives a rationally balanced efficiency. Herein appears the engineering skill and experience required for design of hot water heating.

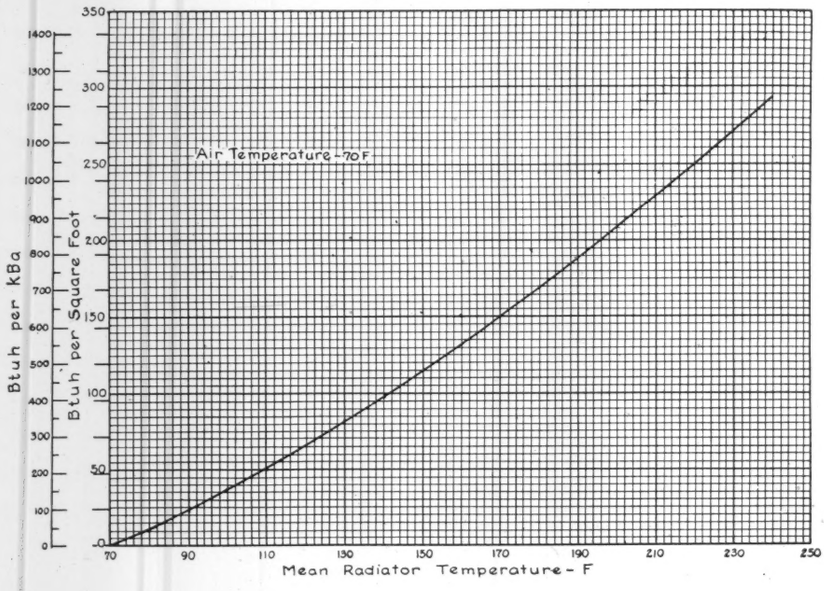
The amount of heat transmitting surface required in any room is a function of the temperature difference between the air in the room and the mean effective temperature of the circulating medium within the radiator. The amount of heat transmitted per unit of radiator area touched by the air varies with the shape and height of the area, and with the speed at which the air passes over it.

In discussing units of heat transmitting surface there always is complication when the obsolescent term square foot is used. One wonders always whether it means some unit which transmits 240 Btu h when in air at 70 F or whether it means the same area transmitting 150 Btu h. The new term *kB* (1000 Btu) is being introduced by the authors coincident with the proposal of the Committee on Nomenclature in its report to the Society. *kBh* means 1000 Btu per hour. *kBa* means that *area* which at 215 deg mean temperature of the heating medium transmits 1000 Btu per hour, in the standard room temperature of 70 F.[1]

When water is used as the heating medium a unit area which would transmit 1000 Btu h (1 *kBh*) when full of steam at 215 F in air at 70 F will transmit less heat or more heat, depending on the water temperature. Throughout this paper, therefore, values will be expressed in both *kB's* and in square feet of radiation, the

[1] At the time this paper was prepared there was no assurance whether *Mb* or *kB* should be the accepted symbol to represent 1000 Btu. The A.S.H.V.E. at its meeting in June, 1933, adopted *Mb* and at its meeting of February, 1934, tabled a resolution to change to *kB*. The authors suggest that the reader, therefore, substitute *Mb* for *kB*.

Fig. 1—Heat transmitted per unit of radiator area



kB's usually being the number of heat units per hour transmitted at the stated water temperatures by the same surface area which would transmit 1 *kB* if the water were at 215 F in thermally circulated air at 70 F. Given a room which loses 10 *kBh* to be heated by a water radiator under conditions in which each unit of surface transmits 0.6 *kBh* there would be required $\frac{10}{0.6}$ or 16.7 *kBa* of radiator surface area.

With a mechanically circulated liquid any two units of area under the same conditions for both will give off heat proportionally as their mean effective temperatures vary.

With hot water the given mean effective temperature may be varied within a very wide range, extending from the room temperature itself well beyond the temperatures of low pressure steam. The heat transmitted to the room per unit of radiator area may likewise be varied in a complete and unbroken range, from no heat at all up to the maximum required. The accompanying chart visualizes the extent of this possible range. It agrees closely with the data given in The A. S. H. V. E. Guide 1933.

It would be possible to change the temperature of the entering water and of the outgoing water within a very wide range indeed without increasing or decreasing the amount of necessary radiating surface, so long as the mean effective temperature is kept constant. Thus for a mean effective temperature of 170 deg, commonly used as a basis for design:

Temperature of Entering Water F	Temperature of Leaving Water F	Temperature Drop in Radiator F	Mean Effective Temperature F
175	165	10	170
180	160	20	170
185	155	30	170
190	150	40	170
200	140	60	170

When steam is used inside a radiator the pumpage phase of the matter is of little moment. The radiator is kept full of steam which condenses to water that occupies only about one-eightieth of the space of the steam from which it was condensed. With a liquid inside the radiator the volume does not change appreciably as the temperature changes. After heating the outside of the radiator nearly to the temperature of the liquid nothing is to be gained by forcing surplus liquid through the radiator. If one pumped enough water through the radiator he could reduce the temperature drop to a very low value and with the same entering temperature could increase slightly the heat transmitted per unit of radiation. The increase, however, would be by no means proportional to the pumpage.

For instance, with a mean effective temperature of 170 F and a 10 deg temperature drop the entering water would be at 175 F and the leaving water would be at 165 F. There would be required under these circumstances for delivery of 0.625 *kBh*, $\frac{625}{10}$ lb of water per hour, that is, 62.5 lb of water per *kBa* of radiator area, or 15 lb of water per square foot of radiator area. With the same *entering* temperature as before, (175 F), doubling the pumpage to a figure of 125 lb of water per hour per *kBa* of radiator surface will raise the mean effective temperature of the radiator only 2.5 deg, and will increase the heat output of the radiator only about 0.020 *kBh* per *kBa* unit area, or 4.75 Btu h square foot. This would be of very doubtful overall advantage, the only compensating factors against the greater cost of pumping being a very slight decrease in the necessary radiator size, and a slightly increased turbulence in the water around the heating surface of the boiler.

On the other hand, if the same *mean* temperature is maintained, that is 170 F, and if the pumpage rate is doubled to 125 lb per hour per *kBa,* the same heat delivery as before will be obtained, namely 0.625 *kBh* per *kBa* unit area, and the only compensating factors against the higher pumping cost will be a slightly reduced temperature of the water around the fire and the same slightly increased turbulence or agitation of the water passing around the heating surface of the boiler.

Experience has demonstrated that in the design of new mechanically circulated hot water systems, using radiation of the same size as is customary with thermal circulation, and a mean effective radiator temperature of 170 F for the most extreme demand, a temperature drop of around 10 deg and a pumpage of about 62.5 lb of water per hour per *kBa,* or 15 lb of water per hour per 150 Btu unit of area, is conservative, and that it will give economical pipe sizes and power consumption.

If it is desired to use radiators no larger than those which would be required with steam, for hot water heating, the water temperature may be increased. Such increase in temperature of the water merely brings down the efficiency of heat transfer from fire to water more nearly to that in effect when steam is used. The saving

in *first* cost due to smaller radiators must then be balanced against the increased *operating* cost due to the lower fuel efficiency, and the engineer must determine to which of these two factors he will give the greater weight. However, due to the very positive and rapid water travel against the heat absorbing surfaces around the fire with mechanical water circulation, it is believed that even at the same temperature differences the efficiency of a water boiler will be higher than that of a steam boiler.

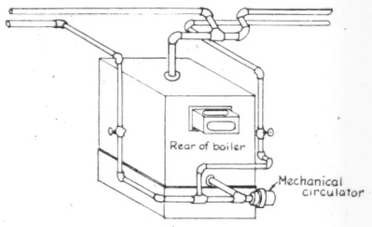


Fig. 2—Typical residence application

The following explains one situation when higher water temperatures are used:

Pumpage per Hour		Ent. Temp.	Temp. Drop	Mean Eff. Temp.	Air Temp.	Conversion Factor	Btuh per Unit of Area	
kBa	Sq Ft						*kBa*	Sq Ft
118 lb	28.4 lb	240 F	10 F	235 F	70 F	1.183	1183	284

Throughout the milder weather which prevails most of the time during each heating season, a mechanical circulating plant will reduce the flow water temperature very greatly without prejudice to the distribution of heat, and will, unlike a thermally circulating plant, maintain this distribution while the flow water temperature is being reduced due to a reduction in the heat demand of the building.

During these hours the boiler water temperature often may be 30 deg or more below that required during those hours when the demand is increasing. The pump insures that all radiators are warm, but the comparatively cool boiler water, as well as a positive movement of the water across the boiler heating surfaces, gives a great increase in the efficiency of heat absorption from the gases of combustion.

This condition is in effect more than half of the time with any well designed mechanically circulated hot water heating system. The condition accounts for fuel savings inherent to a forced hydraulic heating system. For example in one large greenhouse the difference between the out-going water temperature and the return temperature was 90 F with thermal circulation. Thus the mean effective radiation temperature was 45 F below the radiator inlet temperature. Mechanical circulation reduced the 90 F drop to 10 F and maintained the same mean effective temperature with the same heat output, and accomplished saving in fuel which paid for the pump in a few months.

A new modern mechanical circulation piping system is very similar in general layout to a two-pipe vacuum steam system in that each radiator receives a supply connection from the flow system with a return connection to the separate return system. There are no connections from the supply mains to the return mains except through radiators, and the head which causes the water flow, positively is exerted on every radiator.

There are available automatic valves which may be placed on the various main circuits which will throttle the flow due to the pressure in the corresponding return, to which their pressure-sensitive elements are connected. Each automatic valve may be adjusted to permit the existence of the necessary pressure difference and no more, required to overcome the resistance of that particular circuit; thus making any surplus pressure available for use on the less favorable circuits.

The same automatic throttle valves may also be controlled by divisional thermostats so as to reduce the water flow as the room temperature increases. This permits the adjustment of the zone temperature automatically as the sun finds its way around the East, South and West sides of a building. In Northern cities the winter sun at noon looks far into the rooms and its radiant heat warms up the floors and furniture and renders zone control of the type described exceedingly valuable, especially in office buildings.

Some of the older very large hot water heating systems used mechanical circulation for the mains, but depended on thermal circulation for the risers, and had loop connections between these, with many radiators connected into the same riser; the supply connection to the radiator being some feet above the return connection. The pumpage of water necessary to secure results in such systems is enormous, and the temperature drop through any given radiator usually is exceedingly small. Re-arrangement of the piping so as to secure a positive differential is advisable with such old plants.

In general, the velocity of the water through the pumping unit in a new electrically circulated system on the basis of a 10 deg temperature drop should not exceed 2 ft per sec in a plant using a single 2 in. main; 3 ft per sec with the equivalent of a single 4 in. main; 4 ft per sec with the equivalent of a single 6 in. main; 4.5 ft per sec with a single 10 in. main; and 5 ft per sec with a single 12 in. main or its equivalent. The velocities of water listed apply also to the mains in which the units are mounted. Reductions in pipe sizes of mains should be made as the successive radiation load is taken off, and the sizes of risers and radiator branches should be so selected as to maintain the same basic friction loss per foot of pipe throughout the system. Such velocities will work out to a temperature drop of around 10 F in the radiation in the coldest weather, providing that the circulating units are selected as recommended by the manufacturers.

It will be noted that on *old* systems, with all the piping in place and of larger pipe sizes than those suggested, a considerably smaller temperature drop than 10 F usually will be obtained. This will do no harm and will not prejudice the probability of savings.

One objection to hot water heating has been the fact that domestic or service water at adequate temperature has not been obtainable, and separate heaters for this service have been required. These usually burn a concentrated fuel and sometimes require an annual expenditure almost equal to that needed for heating the house.

This matter has recently been solved admirably. The solution requires mechanical circulation but easily may be applied to old thermally circulated systems. A large heat transfer unit is placed near the hot water boiler, receiving the water from the latter around its copper coils. The service water circulates within the coils. The boiler water is maintained at any desired temperature the year around —say 180 F. This easily heats the service water by thermal circulation to say, 160 F.

The hot water mains for the house heating are by-passed off the heater through a non-return valve, and the return from the house heating system passes to the mechanical circulator. The latter operates according to the dictation of a room located thermostat. In summer the mechanical circulator operates not at all, but a very small gas fire or oil fire or stoker fire or manually tended coal fire serves to keep hot the large volume of water in the well insulated boiler. In effect this boiler becomes an enormous thermos bottle.

In one instance out of many of which the authors have knowledge the following tabulation shows the remarkable savings which were effected by this adaption of service water heating. Column 1 gives the cost of heating with a gas burning equipment; Column 2 gives the cost of coal for heating the same demand of service water with the new stoker-fired arrangement:

Month	Column 1 (1932)	Column 2 (1933)
January	$23.00	$2.65
February	22.60	2.70
March	23.40	3.10
April	21.40	1.95
May	20.00	2.65
June	21.00	2.40
July	20.00	2.57
7 Months	$151.40	$18.02

Saving in 7 months. $133.38.

It is admitted, of course, that thus to increase arbitrarily at all times the temperature of the water in the boiler reduces the efficiency of heat transfer, fire to water. On the other hand small direct fired service water heaters in many cases are woefully inefficient. With oil as a fuel in the house heating boiler the service water problem sometimes has been bothersome. The savings to be made by the arrangement described are noticeable under such circumstances.

Some of the ultra-modern convector types of room-located heaters, often placed in a flue, have been found unsatisfactory with thermally circulated hot water. When mechanically circulated positive differential hot water is used these convectors become efficient. Many of the light weight convectors are so quick to heat and to cool when steam or vapor is used that they are considered unsatisfactory. The use of mechanically circulated water with its factor of stored heat often removes the objection.

Water boilers, whether under thermal or mechanical circulation, have cracked sections or burned crown sheets or explosions very rarely indeed. The water level in the system can vary a great many feet without danger of damage.

Old thermally circulated hot water heating systems easily have been changed to two-pipe steam systems. It is interesting to know that there are a number of two-pipe systems with the small return pipe sizes satisfactory for vacuum pump return which have been changed to mechanically circulated water systems. There will be more of these. All that is necessary to do is to remove the traps from the radiators and the vacuum specialties from the boiler and to provide means for air venting and expansion. The pump may have an unusually high resistance to overcome with such small return piping, but usually will require less energy than did the vacuum pump or condensation pump which may have preceded it in service.

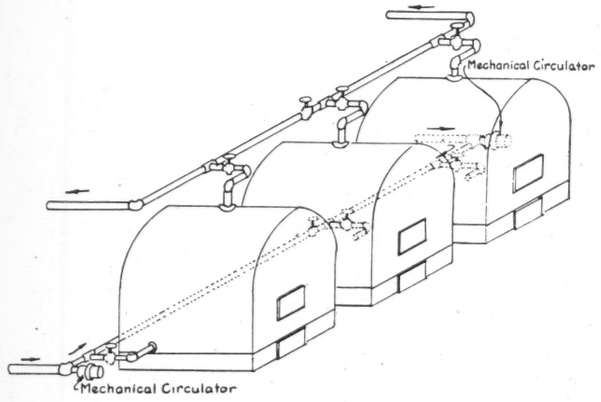


Fig. 3—Typical mechanical circulators as used for large greenhouses

The mains are often as large as 12-in. diameter

A mechanically circulated hot water system may be used as a summer cooling system if a supply of chilled water is available. Such a plant utilizes a maximum part of the investment the year around. The radiators used for cooling must have drain pans under them and should have fans under or near them to induce rapid air circulation.

The same radiators as are used for hot water heating will accomplish a very substantial amount of cooling under such circumstances. The insulation used on the pipes and surfaces of such a plant should be inert to the moisture which will condense out of the surrounding air. Such insulation, for heating and for cooling, is available in the market.

Conclusions

Heat transfer by forced liquid which is forced to circulate independently of thermal conditions has many advantages.

The value of having a variable mean effective temperature of the heat transmitting medium cannot be over-emphasized. Savings in fuel and in comfort invariably attend heating systems which adjust the mean effective temperature of the heating medium to suit the varying demand. Such systems reduce greatly the difficulty of temperature control.

They lend themselves admirably for adaption to cooling as well as to heating and are capable of providing service hot water as effectively as do steam heating plants.

Old liquid heating systems not having pumps can be improved and corrected by installing modern electric circulators.

A Proving Home for Air Conditioning Investigations

By **Elliott Harrington,[1] *(MEMBER)* and Leon A. Mears,[2] *(NON-MEMBER)***

Schenectady, N. Y.

THIS is the first of a series of papers to be presented from time to time, on the investigations under actual living conditions at the Proving Home of the General Electric Co. This air conditioned home was unofficially established in September, 1928, for the purpose of testing the oil furnace under actual field conditions. On March 1, 1933, it received official recognition as part of the General Electric Air Conditioning Institute and was set up to accomplish the following four-fold objective:

1. Improvement of air conditioning apparatus, both in production and in development.
2. Improvement and simplification of installation and application standards.
3. Accumulation of additional data on the benefits of air conditioning to comfort and health.
4. General public education on matters pertaining to air conditioning, through the publishing of test reports and through public demonstrations.

General Description

The Proving Home is located in Schenectady, N. Y. This location affords a study of operations under nearly all the climatic conditions met with in this country. Summer and winter extremes of 104 deg above and 24 deg below zero, respectively, have been recorded. The mild spring and fall of this locality round out the picture.

In order to demonstrate that year 'round air conditioning can be applied to the average house, the Proving Home is of ordinary frame construction as indicated by Fig. 1. The only special provisions are the addition of storm sash to all windows, awnings on the sun porch windows for special testing and ½ in. of rigid board insulation on the under side of the roof rafters. Windows are of the usual double-hung construction without weather strip. Doors are also without weather strip and are equipped with combination screen and storm doors. Outside walls are made up of wood shingles, building paper, ship lap, studding, wood lath and plaster.

Fig. 2 shows the plan of the first, second and third floors, as well as the duct layout and location of all registers and radiators. It will be seen that the various types of rooms which go to make up the problem of air conditioning a home, are well represented in this layout, as follows:

1—Normal first floor room over heated area (Living Room).
2—Normal second floor room over heated area (Bedroom 1).

Fig. 1—The proving home in winter

3—Normal first floor room over unheated area (Library).
4—Room with large glass exposure over unheated area (Sun Porch).
5—Normal third floor room (Maid's Room).
6—Built-on garage (directly back of Library—not shown).

All rooms of the home are completely furnished in the usual manner, including a fully equipped laundry, recreation room and boys' shop in the basement, as well as the machine room and instrument room.

In spite of the reliable application engineering information available, it is believed that in the present state of the air conditioning art, the data furnished by the various indicating and recording instruments must be supplemented by the actual physical reactions of various people living in the home. Therefore, two elderly people, a middle-aged man, a young man and two children live in the Proving Home at all times.

Table 1 shows the calculated heat gain and heat loss of each room, and Table 2 summarizes these calculations.

Distribution System

In addition to facilities for installing and testing various types of unit apparatus, two central distribution systems are installed as follows:

1. One pipe steam with basement loop, and wet return.
2. High volume, low pressure duct system.

By means of valves and double throw switches, either system may be used, independently of the other.

The radiator system includes standard height, window height, baseboard and enclosed radiators for study purposes. Vacuum vent valves are used throughout. The radiator in the sun porch is equipped to operate as a two pipe radiator, in order to study the operation of radiator thermostatic valves.

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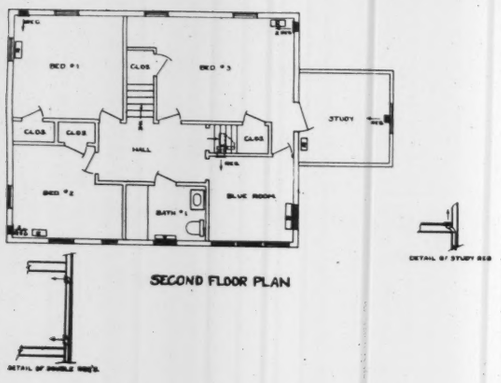
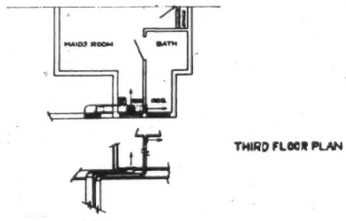
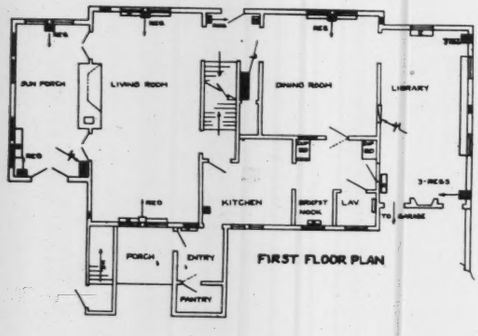
[2]Air Conditioning Dept., General Electric Co.

Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.







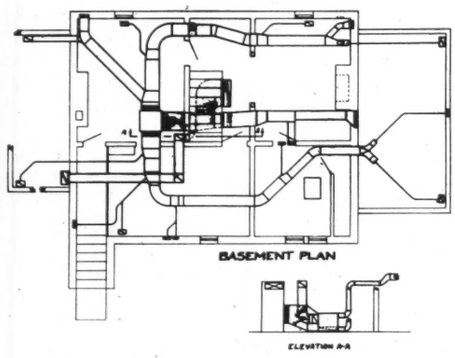


Fig. 2—Floor plan and duct layout

The duct system is laid out as shown in Fig. 2, and includes the following special features:

1. Four floor returns—one each in the front hall, library, sun porch and third floor stair well.
2. One horizontal delivery ceiling return in the library.
3. Six delivery openings in the library—two in the floor discharging vertically and two each at the baseboard and close to the ceiling, discharging horizontally.
4. Horizontal Ceiling and horizontal floor delivery in Bedrooms 1, 2 and 3.

It will be seen that the library can be conditioned with almost any conceivable type of air flow by using the re-

Table 1—Heat Loss and Heat Gain Calculations for Proving Home

Room	Volume Cu Ft	Heat Loss		Heat Gain		Radiation Sq Ft E. D. R.	
		Btu per Hr	Heated Air CFM	Btu per Hr	Cooled Air CFM	Calculated	Installed†
Sun Porch	1020	14080	216	5040	137	58.5	34
Living Room	2450	7820	120	4390	120	32.5	80
Hall	606	2690	41	830	23	11.0	20
Dining Room	1400	4660	71	2930	80	19.5	40
Library	1650	15980	245	4710	128	66.5	62
Lavatory	156	1500	....	Not	Cooled	6.0	5
Breakfast Nook	156	1500	....	Not	Cooled	6.0	5
Kitchen	867	3400	....	Not	Cooled	14.0	15
Entry	260	4710	....	Not	Cooled	19.5	18
Bedroom No. 1	1080	7820	138	3420	107	32.5	20
Bedroom No. 2	886	5850	103	3040	95	24.5	30
Bedroom No. 3	1670	9240	163	3470	109	38.5	40
Blue Room	720	5610	99	2360	74	23.5	38
Bath	394	2130	....	Not	Cooled	9.0	18
Study	600	4110	72	3510	110	17.0	30
Maid's Room	430	3790	78	3010	110	16.0	30
Maid's Bath	250	2610	54	1540	57	11.0	...
Garage	1900	9600	....	Not	Cooled	40.0	40
Totals	16495	107100	1400	38250	1150	445.5	525

†This radiation was installed when the house was built in 1924. It indicates how inaccurate "Rule-of-Thumb" methods can be.

Assumptions

1. *Heating*
   (a) Lowest recorded temperature −24 F.
   (b) Design temperature −9 F.
   (c) Indoor temperature 70 F.
   (d) Calculated Relative Humidity, 42 per cent.
   (e) Heated volume, exclusive of garage, 14,595 cfm.
   (f) Degree days in a normal heating season of 8 months, based on 65 F, 6889.
2. *Cooling*
   (a) Outdoor design dry bulb temperature 91 F.
   (b) Outdoor design wet bulb temperature 74 F.
   (c) Indoor dry bulb temperature 78 F.
   (d) Moisture gain 0.17 lb per hour per person.
   (e) Fresh air supply of 300 cfm definitely provided by kitchen exhaust fan. This corresponds to approximately 1¼ changes per hour for entire house.
   (f) Awnings over windows exposed to sun, and attic ventilator are used.
   (g) Cooled volume 12,760 cfm.
   (h) Degree hours in normal cooling season of 4 months based on 12 hours of cooling per day, 7310.
3. *Circulation*
   The cfm requirements for air conditioning were obtained by proportioning the total cfm of the air conditioner according to heat loss or heat gain of room after correcting the Btu requirements to allow for duct losses. 15% was added to Btu of second floor rooms and 35% to third floor rooms. Air conditioner to deliver 1400 cfm of heated air. Same conditioner will deliver 1150 cfm of cooled air.

Table 2—Summary of Heat Loss and Heat Gain Calculations for Proving Home

Heat Loss in Btu Per Hour	
Infiltration	16,000
Conduction through walls	43,900
Conduction through windows and doors	9,000
Conduction through roof	16,000
Conduction through floor over unexcavated basement	4,700
Over-all Exposure Factor 1.09	89,600
Total heat loss above grade	97,500
Basement Loss	13,200
Garage—(50 F)	9,600
TOTAL HEAT LOSS OF HOUSE	120,300
Heat Gain in Btu Per Hour	
Walls	
Conduction due to temperature difference	9,600
Additional conduction due to sun effect	9,000
Windows	
Conduction due to temperature difference	1,900
Direct sun effect	17,800
†Roof	
Conduction due to temperature difference	3,100
Additional conduction due to sun effect	10,200
Fresh Air Supply (300 cfm)	12,600
Occupancy (5 persons)	2,000
TOTAL HEAT GAIN WITHOUT ATTIC VENTILATION AND WITHOUT AWNINGS	66,200
TOTAL HEAT GAIN WITH ATTIC VENTILATION AND WITHOUT AWNINGS	56,000
TOTAL HEAT GAIN WITH AWNINGS AND WITHOUT ATTIC VENTILATION	48,400
TOTAL HEAT GAIN WITH ATTIC VENTILATION AND WITH AWNINGS	38,200

†NOTE: The conduction through roof due to temperature difference is calculated heat gain through ceiling with an attic temperature of 96 F, and is the gain when an attic ventilator of sufficient capacity to maintain this attic temperature is used. The additional conduction due to sun effect is increase of ceiling conduction due to high attic temperature without attic ventilator.

quired combination of supply openings, return openings, delivered air temperatures and air volumes. Ordinary cross-bar grilles are used. Ducts are insulated with 4-ply cellular asbestos, which is moisture proofed and sealed. The entire duct system was installed after the house was built, thus showing this procedure to be practical.

The kitchen, garage and bathrooms are heated by steam radiation to avoid recirculation of odors. Also to prevent an undue concentration of cooking odors and to give a measure of temperature control in the summer, the kitchen is forced ventilated to the out-of-doors. Figure 3 shows a typical delivery duct and radiator installation in the library.

Mechanical Equipment

The permanent air conditioning equipment installed in the home consists of:

1. *Oil Furnace*—133,000 Btu/hr maximum output. (Referring to Table 3, it will be seen that the oil furnace is too small for this application, under normal operation. However, the application was made in this manner for the purpose of studying methods of reducing "pick-up" load, as will be explained in a later report.)
2. Central air conditioner 130,000 Btu/hr maximum sensible heat output with 1,600 cfm air circulation and a humidifying capacity of 1.5 gal of water per hour (as installed at the Proving Home, the heat output has been reduced to 115,000 Btu/hr with 1400 cfm air circulation).
3. Cooling attachment to the central air conditioner with a heat absorbing capacity of 44,000 Btu/hr and a dehumidifying capacity of 1.5 gal of water per hour.
4. Two refrigerating units, each of 24,000 Btu/hr capacity.
5. Attic ventilating fan of 1,600 cfm capacity.

The oil furnace generates steam between the limits of 0.5 and 6 lb steam pressure to heat the transfer surface in the air conditioner whenever the room thermostat calls for heat. The oil furnace also heats domestic hot water the year 'round, by means of an indirect heater. Further, it is used to supply steam for the radiation heating system.

The air conditioner draws air through the return system, over the cooling and dehumidifying surface and through the metal wool filters, delivering the air over the heating and humidifying surfaces to the delivery duct system.

The refrigerating units operate on a call from the room thermostat for cooling, to cool the extended surfaces at the return entry to the air conditioner. Either direct or indirect expansion is available for testing purposes.

Fig. 4 is a general view of the machine room, showing the oil furnace and refrigerating units and Fig. 5 shows the air conditioner with cooling attachment.

Fig. 3—Delivery duct and radiator installation in the library, showing thermo-couples to measure air inlet temperatures

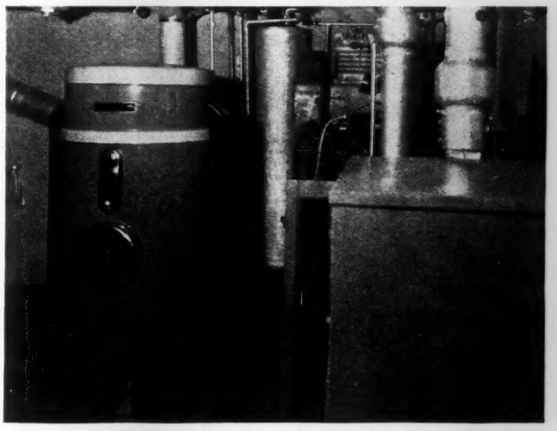


Fig. 4—Machine room, with oil furnace and refrigerating units

The attic ventilating fan is arranged to take air in at one end of the attic and expel it at the other end, whenever the attic temperature is higher than the temperature out-of-doors. It is installed so it can also be used during the night to draw cool night air up through the house and out through the attic.

Control System

The control system is made up as follows:

1. An electric driven time thermostat located in the dining room operates to control the starting and stopping of the oil

furnace as required for temperature control, when using the radiation heating system. (To date, no humidity control has been provided with the radiation heating.) This same thermostat controls winter temperatures with the air conditioning system by starting and stopping the blower and oil furnace as required. It also controls summer temperatures by starting and stopping one or both of the refrigerating units, depending upon the cooling load The capacity of the refrigerating units is such that the maximum dry bulb cooling which can be obtained is about 15 F.

2. Humidifying is controlled by a humidistat located in the dining room and operating a solenoid valve in the water supply

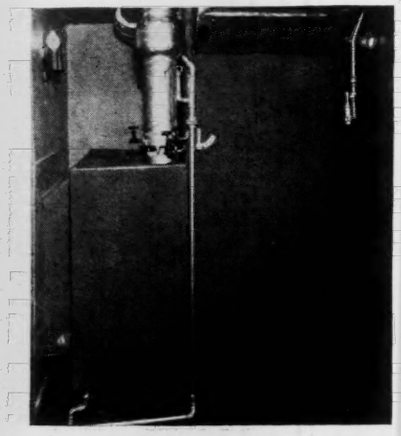


Fig. 5—Central air conditioner with cooling attachment

line to the humidifier. Dehumidification is inherently controlled by the predetermined temperature of the cooling surface..

3. The radiation in the kitchen and bathrooms is controlled by an independent room thermostat operating a motor-valve in the steam supply line, regardless of whether the radiation or conditioning system is being used.

4. The garage radiator is controlled by another independent room thermostat and motor valve combination, so interlocked that the valve can only admit steam to the radiator when the garage doors are closed.

Measuring Equipment

Temperatures are measured by 102 thermocouples mounted at various levels and in various locations in each room as well as in the attic spaces, basement spaces, out-of-doors, and in certain parts of the machine equipment and duct system. These thermocouples are read on a potentiometer through a selector switch system and also may be automatically recorded on a recording potentiometer. Temperatures are also continuously read and recorded on three day or seven day charts, in the dining room, library, sun porch, garage, kitchen, bedroom No. 1 and maid's room, as well as out-of-doors.

Humidity is recorded continuously both inside and out, the recording instruments being regularly checked with a sling psychrometer. Humidity distribution is so even that it is felt unnecessary to record it at more than one place.

Air velocities are measured by sensitive electric hot wire anemometers. All suspensoids collected by the filters are analyzed for composition and for size of particles. A dust counter is used to determine the dust content of the air both inside and out.

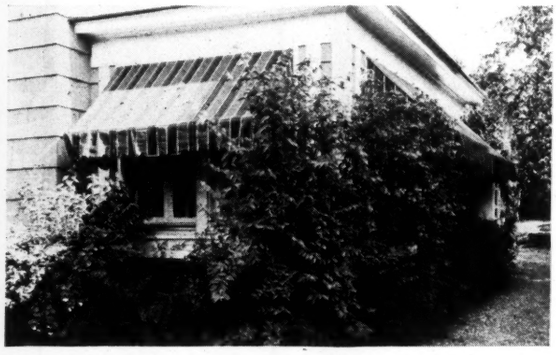


Fig. 6—Awning installation on sun porch

Operating time is totalized for each operating machine, using individual electric time meters. In addition, on-off operating cycles are recorded on a chart, for each operating machine.

Electric power consumption is totalized for each machine on individual integrating watthour meters.

Oil consumption is read periodically from a calibrated direct reading meter.

Water consumption is totalized for each machine on integrating meters.

Wind velocities are obtained from the weather bureau.

All instruments are calibrated by and under the supervision of the General Engineering Laboratory.

Special Equipment

1. The sun porch is equipped with awnings and this room is used particularly to study heat gain calculations. (Fig. 6.)

2. A cooling tower is available and it is planned to conduct a series of tests during the summer of 1934,

Table 3—Boiler Load for Proving Home

Air Conditioning		Steam Heating	
[1]Air Conditioner Rating	115,000	Direct Radiation in House (485 sq ft)	116,400
[1]Direct Radiation in Spaces not Air Conditioned (61 sq ft)	14,650	Direct Radiation in Garage (40 sq ft)	9,600
[1]Direct Radiation in Garage (40 sq ft)	9,600	Pipe Tax (14% of 525 sq ft)	17,600
[2]Pipe Tax (14% of 101 sq ft)	3,400	Pick Up (25% of 525 sq ft)	31,500
[3]Pick Up	.....	[4]Domestic Hot Water	.....
Domestic Hot Water	12,500		
Total Boiler Load	155,150 Btu/hr	Total Boiler Load	175,100 Btu/hr

Notes:
[1]The air conditioner *rating* for the blower speed employed is used here; not the heat loss of the air conditioned space, since the former is the actual load imposed on the boiler during periods of operation. For the same reason, *installed* radiation is used, and not calculated.
[2]One pipe steam system with all mains, insulated.
[3]No "pick-up" allowance for split system since direct radiation is only 21% of air conditioner rating.
[4]Domestic hot water neglected in steam system since it is only 40% of "pick-up" allowance.

Table 4—Normal Winter Run

Nov. 15—Dec. 13, 1933

Type of Heating, Etc.	Dates of Test	Time in Hours	Oil Consumption (Gallons)	Power Consumption (KWH)	Degree Days (65 F Base)	Oil Cost Per Normal Season of (6889 Deg Days)	Power Cost Per Normal Season of (6889 Deg Days)	Total Operating Cost per Normal Season of (6889 Deg Days)
Steam	(11-30-33) to (12-4-33)	93.75	41	7.8	123.3	$137.90	$13.00	$150.90
Complete Air Conditioning	(11-15-33) to (11-30-33)	354.75	200	64.5	505.8	163.50	26.40	189.90
Complete Air Conditioning	(12-4-33) to (12-13-33)	217.0	143	46.7	369.8	159.90	26.10	186.00

Oil Furnace Total Operating Time (Mins.)	Air Conditioner Total Operating Time (Mins.)	Operating Time of Air Conditioner in % of Total Duration of Test	Operating Time of Oil Furnace in % of Total Duration of Test
2004			35.7
9438	8525	40	44.4
6980	6261	48	53.6

Note: Degree days figured from Proving Home temperature charts, and based on average hourly temperatures.

to determine the possibilities of such a device for condenser water cooling, in homes.

3. A recording ion counter has been developed by the Research Laboratory and is installed in the library of the Proving Home. A study of the ion count with various degrees of habitation and various climatic conditions, is being carried out.

Test Results

An intensive schedule of investigation work is under way, covering a period of at least two years. As stated at the beginning of this paper, it is planned to present certain of the data obtained in the Proving Home, at regular intervals. Thirty-one separate investigations are under way or have been completed to date. The results of four of these investigations are presented here with the hope that they may be of interest and help to the air conditioning industry.

I. General Winter Operation Test

From November 15 to December 13, 1933, the air conditioning system at the Proving Home was operated normally by the regular automatic controls. An exception to the above was taken during the four day interval when the air conditioning system was shut down and the house heated entirely by steam radiation. This latter was done to provide a comparison between steam radiator heating and complete air conditioning. In order to insure a true picture of the operation, the engineer supervising the tests did not manipulate the apparatus or automatic controls in any way. His entire time was spent in recording all possible data. Following is an outline of the records kept:

1. Temperature on all regular indoor and outdoor recording thermometers.
2. Humidity recorded indoors (1st floor), outdoors, and in the garage; with sling psychrometer readings taken on the second floor and on the first floor to check the recording instruments.
3. Operating cycles recorded for the Oil Furnace and Air Conditioner.
4. Power consumption of all equipment.
5. Daily domestic hot water consumption.
6. Total operating time of all equipment.
7. Oil consumption.
8. Dust accumulation in living quarters with dust count readings taken at intervals, both indoors and out.
9. Condensation conditions on all windows.
10. Temperatures between storm sash and inside windows.
11. Quantity of hot water stored in domestic hot water tank at frequent intervals.
12. Outside weather conditions.
13. Special tests. Some of these will be summarized later in this report.

Table 4 gives a brief summary of the operation.

The operating costs were calculated on the basis of electric power costs at 3c per kwh and No. 4 oil at 6c per gallon. The degree days were figured on a 65 F basis. In order to obtain the cost of operation for a normal heating season, the cost for the period of the test was multiplied by the ratio of the degree days in an average Schenectady heating season (6889) to the degree days during the test.

The heating costs for steam are slightly lower than they should be. As the oil furnace is purposely overloaded, about 30 per cent, two or three of the radiators at the end of the one pipe steam system were never entirely hot and the heating of these rooms suffered proportionately. A previous check at the Proving Home, over a complete heating season and with a similar type of steam heating equipment of adequate capacity, showed a total heating cost for a normal season of $159.00. The overload on the furnace is not so apparent, when using the air conditioning system, because of even heat distribution and, as a result, these air conditioning costs are reasonably accurate.

A 60 deg temperature was maintained at night, which accounts for the low percentage of the total "time-on" for the air conditioner.

During the test the air conditioner filters collected about one ounce of dust. This dust collection is less than would be expected. It can be accounted for by the fact that the dust count in the Proving Home had been reduced to below 30 particles per cubic centimeter, with the out-of-door count averaging less than 50 particles per cubic centimeter during the test. The collected dust was very fine, some of it being but one micron in diameter.

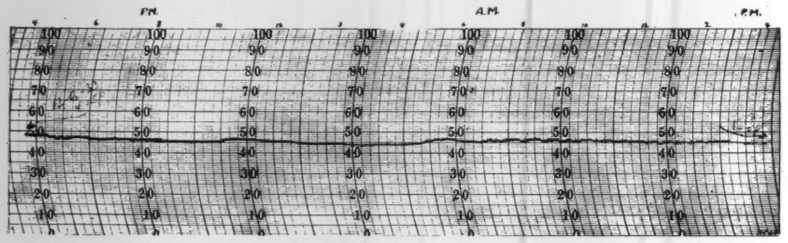


Fig. 7—Relative humidity chart for one day during normal winter run

The following outline indicates the advantages and disadvantages of the complete winter air conditioning over heating the house by radiation, as shown by this test.

Advantages

1. Substantial reduction of dust accumulation in the living quarters. (The maid found it necessary to dust only every third day instead of every day, as with radiation heating.)
2. Humidification under positive control. (See Fig. 7 for the record of humidity control for a typical day at the Proving Home during the test.)
3. Closer temperature control. (See Fig. 9 for temperature record in Dining Room with steam heating and with air conditioning.)
4. Uniform temperature more readily obtained throughout house. (See Table 5, for typical temperatures taken almost simultaneously. See Figures 8 and 10 for 3 day temperature records in Dining Room, Library, and Second Floor Bedroom.)
5. Floor to ceiling temperature differential decreased. (See Test No. III.)
6. Elimination of normal tobacco smoke odors. This occurs since the smoke is diluted and distributed through the house and its absorption into walls and solid materials is increased by the greater area available and by the flocculation accomplished by the air circulation.

Disadvantages

1. Operating cost approximately 15 per cent greater than for steam heating. (This is not a general conclusion. *For equal comfort,* the operating cost of the air conditioning system should not exceed the operating cost of the radiation system by more than the extra power required for air circulation. The excess cost may be less than this due to lower mean temperatures for the same degree of comfort.)
2. Higher first cost in remodeled heating systems.

Table 5—Breathing Line Temperatures of Individual Rooms in Degrees Fahr with Complete Air Conditioning System in Operation, After Reaching Steady State Conditions

3rd Floor Bath	75	2nd Floor Study	74
3rd Floor Bedroom	74	1st Floor Hall	74
2nd Floor Bedroom No. 1	73	1st Floor Living Room	75
2nd Floor Bedroom No. 2	73½	1st Floor Library	78½
2nd Floor Bedroom No. 3	74	1st Floor Dining Room	74
2nd Floor Hall	74	1st Floor Sun Porch	75½
2nd Floor Blue Room	74		

1st floor library maintained at higher temperature because of cold floor conditions. (Room over un-excavated space).
Controlling thermostat set at 74 F.
Temperatures measured at center of room, at the breathing line.

II. Domestic Hot Water Consumption

During the normal winter tests, daily hot water consumption was recorded. (See Figs. 11 and 12.) The occupancy of the house during this period was three adults, one child, and one maid. The household was not informed of the recording in order to keep the water consumption absolutely normal. The 24-hour periods are from 4:00 p. m. of one day to 4:00 p. m. of the next. This time was chosen as it was inconvenient to record from midnight to midnight. Table 6 summarizes the daily hot water consumption records.

The outlet of the hot water storage tank is connected through a thermostatic mixing valve, so that the outlet water temperature is held constant at 125 F plus or minus about 2 F.

Table 6—Domestic Hot Water Consumption

Duration of Test (Hours)	665.5
Total Hot Water Used (Gallons)	1,795
Average Daily Consumption (Gallons)	64.7
Maximum Daily Consumption (Gallons)	105.5
Minimum Daily Consumption (Gallons)	30.5
Temperature of Hot Water (F)	125
Average Wash Day Consumption (Gallons)	95.6
Average Daily Consumption Excluding Wash Day	60.6

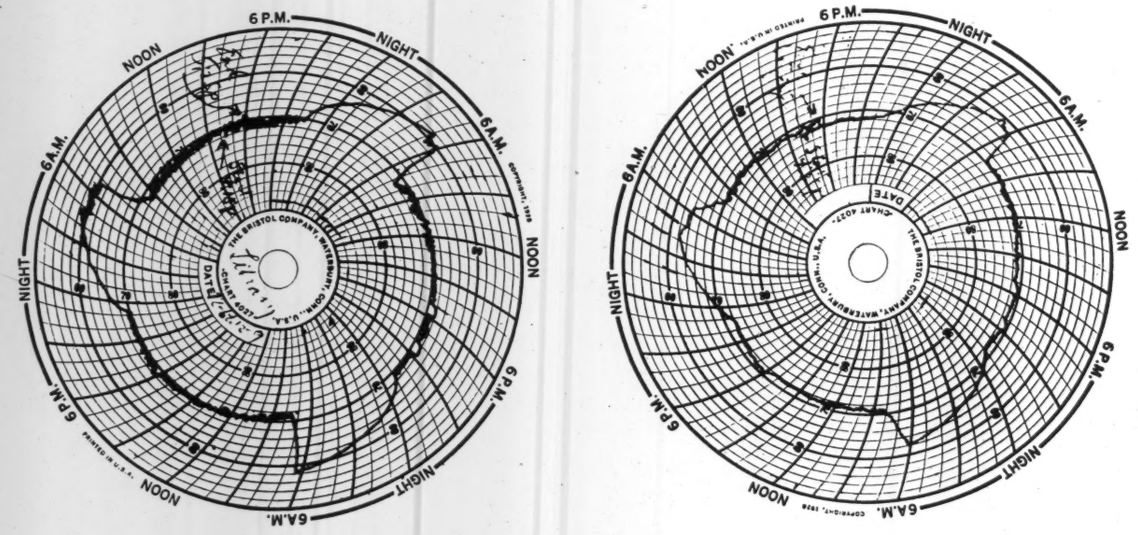


Fig. 8—Typical dry-bulb temperature charts from library and bedroom No. 3, showing temperature balance in various parts of the home with air conditioning. Controlling thermostat in dining room set to hold 71 F

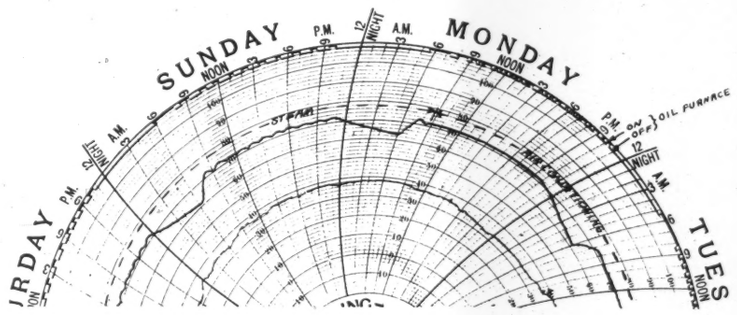


Fig. 9—Comparison of dining room temperature maintained with steam heating and with air conditioning. Heavy line shows dining room temperature. Light line shows outside temperature

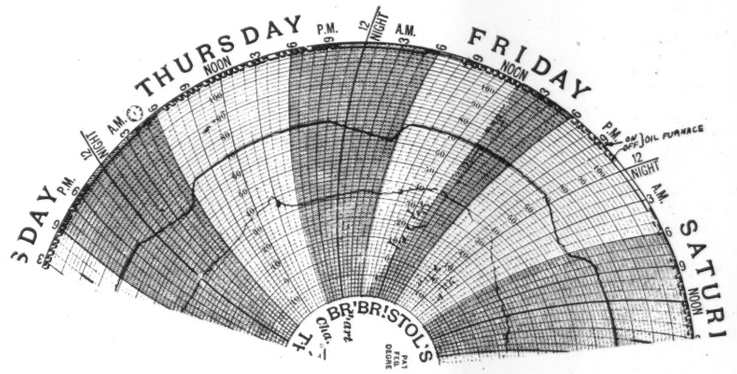


Fig. 10—Dining room temperature with air conditioning and widely varying out-of-door temperature. Same period of time as shown in Fig. 8. Heavy line shows dining room temperature. Light line shows outside temperature

III. Temperature Differential Between Floor and Ceiling

In recording floor to ceiling temperature differentials (hereafter referred to as differential), the air was supplied to the rooms at temperatures varying from 90 to 135 deg. The following factors were found to affect the differential and with each test, mention will be made as to how certain of these factors were controlled.

1. Blower operating time, prior to reading temperatures.
2. Temperature of air delivered to room.
3. Outside temperature.
4. Rate of air delivery.
5. Location of supply and return grilles.
6. Type of heating system.
7. Construction of room.

It was found that if air of a constant temperature and of sufficient volume to provide about six air changes per hour was delivered to a room, the differential became practically constant in 2 to 3 min.

Further, it was found necessary to be able to measure the temperatures with thermocouples. Temperature changes were effected within a few seconds and as a result it was possible to get true readings, only with a thermocouple of small mass, which followed the air changes closely. All differentials were read at the center of the room.

Test "A"

Maximum and minimum differentials were first obtained under normal operation with steam heating and with air conditioning. Table 7 summarizes the most pertinent data obtained.

In this test, readings were taken every half hour for 8 hours during the day, but did not include the long morning pick-up heating cycle. The readings were taken without setting up special conditions, so they cannot be compared too closely. However, considering the number of readings, it is believed that typical operating results were obtained. The lower horizontal supply and return grilles in the library and the lower horizontal supplies on the second floor were used. Other supplies were all vertical from the floor. Readings were taken for the air conditioning system, whether the blower was operating or not, but a sufficient number of readings were taken with the blower operating to get representative differential readings. Outside temperature conditions varied throughout but are given in Table 7, with the other tabulations.

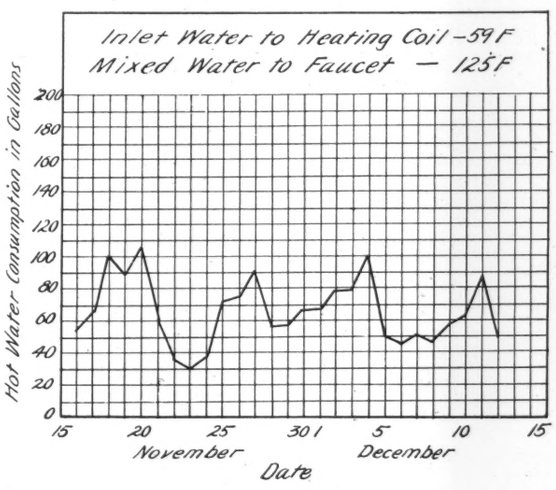


Fig. 11—Daily domestic hot water consumption

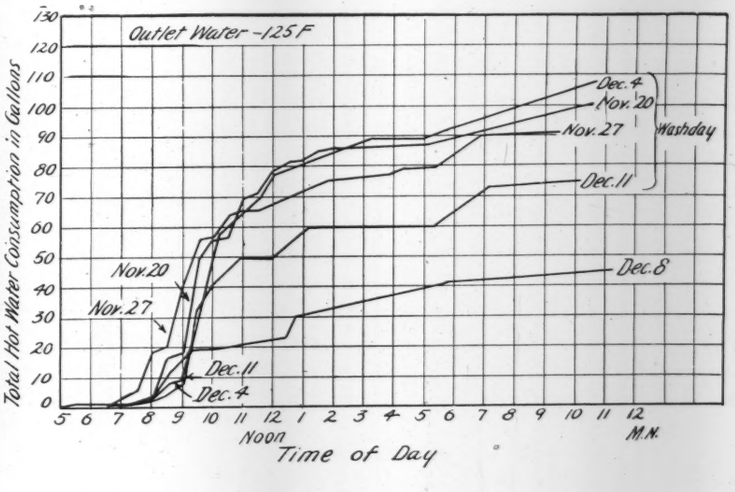


Fig. 12—Domestic hot water consumption

Table 7—Maximum and Minimum Floor to Ceiling Temperature Differentials with Steam Heating and with Air Conditioning under Normal Operating Conditions

Room	Air Conditioning								Steam Heating		
	Differentials		Air Flow CFM	Outside Temp. F	Differentials		Air Flow CFM	Outside Temp. F	Differentials		Outside Temp. F
	Max. F	Min. F			Max. F	Min. F			Max. F	Min. F	
Sun Porch (unheated floor)	11¼	8	150	24° to 12°	7	3½	150	46° to 44°	14	11¾	34° to 28°
Living Room	6	3¼	65		3½	1¾	65		8¼	5¼	
Bedroom No. 3	11	7	80		9	4½	80		14½	8¾	
Library (unheated floor)	15½	8	140		9½	3½	140		17½	9¾	

(This table does not include differentials during long morning pick-up heating cycle).
Temperature of air supplied to rooms varies between 90 and 135 F.

Test "B"

Maximum differentials with various combinations of supply and return grilles in the library, were studied.

In this test the outside temperature remained practically constant. The delivery air temperatures varied slightly and enough to need correction. The rate of air delivery was held constant and the supply air temperature was held constant long enough to produce a steady state condition. Table 8, gives the summary with the duct combinations referred to.

Table 8—Maximum Floor to Ceiling and Breathing Line to Ceiling Temperature Differentials in Library with Different Combinations of Supply and Return Grilles

(After Stable Condition Reached for Given Supply Air Temperature)

Combinations of Supply and Return Grilles Used†	Maximum Floor to Ceiling Differential F	Maximum Breathing Line to Ceiling Differential F	CFM Air Delivery	Outside Temp. F	Supply Air Temp. to Room F
1	11.5	1.9	240	26	113
2	25.5	11.5	240	28	118.5
3	24.9	12.9	240	28	114
4	12.5	1.8	240	29	123
4	14.5	2.0	140	24	120
5	26.5	7.2	240	29	122
Steam Heat	21.5	4.0	...	38	.....

†Key to grille combinations:
1. Lower Horizontal Supply—Upper & Lower Horizontal Return.
2. Upper Horizontal Supply—Upper & Lower Horizontal Return.
3. Upper Horizontal Supply—Upper Horizontal Return.
4. Lower Horizontal Supply—Lower Horizontal Return.
5. Upper Horizontal Supply—Lower Horizontal Return.
6. Lower Horizontal Supply—Upper Horizontal Return.
7. Lower Vertical Supply—Lower Horizontal Return.
8. Lower Vertical Supply—Upper Horizontal Return.
9. Lower Vertical Supply—Upper & Lower Horizontal Return.

Test "C"

Differentials with various combinations of supply and return grilles in the library, taking readings 10 minutes after the blower started operating, were obtained.

In this test the blower started to deliver air at about 90 F and the temperature gradually rose during this test. (This is typical of one cycle of the air conditioner operation.) The air delivery, outside temperature and time of taking readings were constant throughout. See Table 9, for details.

Test "D"

Differentials with steam heating and with air conditioning after a prolonged heating cycle and a constant differential had been reached, were obtained.

This test was taken during the morning heating-up cycle, so that a long period of heating could be obtained. It will be noticed that the outside temperature was much higher for the steam heating test than for the Air Conditioning Test, which factor indicates that the contrast between the two cases would be more pronounced than the figures indicated. In the library the lower horizontal supply grille was used while in the other rooms a lower vertical supply was made use of. Table 10, gives the tabulation.

Table 9—Floor to Ceiling and Breathing Line to Ceiling Temperature Differentials in Library, with Different Combinations of Supply and Return Grilles

Combinations of Supply and Return Grilles Used	Floor to Ceiling Differential F	Breathing Line to Ceiling Differential F	CFM Air Delivery	Outside Temp. F	Supply Air Temp. at Time of Reading F
1	9½	3½	140	33	104
3	12½	10	140	31	95
4	8½	2¼	140	30	100
5	14	7	140	33	98
6	8½	3½	140	32	113
7	10¼	2	140	34	102
8	12½	2¼	140	32	113
9	11½	2	140	32	101

Note: For key to grille combinations, see Table 8.

Table 10—Floor to Ceiling Temperature Differentials After a Prolonged Heating Cycle and a Constant Differential Had Been Obtained

	Sun Porch	Library	Living Room
Steam Differential, F	12	21½	9
Outside Temperature, F	38	38	38
Air Conditioning Differential	11	14½	5
Outside Temperature, F	24	24	24
Supply Air Temperature, F	125	120	125
Cfm Air Delivery	150	140	65

Conclusions

1. The air conditioning system reduced the differential of steam heating from 25 to 50 per cent.
2. The lower horizontal supply with a lower horizontal return gives the lowest differential, with a given air delivery temperature and volume, for winter air conditioning.
3. The living room presents an ideal condition for low differentials with a warm floor, due to heated space directly underneath.
4. The library with cold floors, due to no excavation underneath, presents the worst condition met with and future work will be carried on as to the best methods of remedying such a condition.

IV. Summer Air Conditioning

The summer air conditioning equipment was not completely installed in the Proving Home until late in the summer. As a result, most of the time during warm weather was spent in testing the apparatus itself. However, with the apparatus ready for operation, a comprehensive program has been outlined for the summer of 1934.

From the limited data obtained, it was possible to calculate the power cost for an average Schenectady summer season of 4 months and 7310 degree hours, based on 78 F indoor dry bulb temperature. This figure was $116 based on 3c power, but it is subject to further checking before acceptance.

From observations made on ceiling to floor temperature differential in the summer, the maximum differential recorded was 5 deg, based on horizontal delivery just below the ceiling, and floor return. Consequently, the floor to ceiling differential probably does not hold the importance in summer air conditioning that it does in winter air conditioning.

A. C. Willard, *President*
University of Illinois

A. C. Willard President of University of Illinois

AT A meeting of the Board of Trustees on March 13, Arthur Cutts Willard was selected president of the University of Illinois and will take office July 1. The action of the Board was unanimous and President Willard succeeds Harry Woodburn Chase, who became Chancellor of New York University last year. President Willard has been Acting Dean of the College of Engineering for some time and as head of the Department of Mechanical Engineering he received world-wide recognition for his research work. He is an internationally known authority and is an outstanding engineer, author, educator and research investigator.

President Willard was born in Washington, D. C., and after receiving his preparatory education in the public schools of the District of Columbia, he attended Massachusetts Institute of Technology and was graduated in 1904. He commenced his teaching career immediately in California and in 1906 became assistant professor of mechanical engineering at George Washington University. From 1909 to 1913, he served the Government as sanitary and heating engineer, in the office of the Chief Quartermaster of the Army.

Since 1913, President Willard has been a member of the faculty of the University of Illinois, first as assistant professor of heating and ventilation, and in 1917 he became professor of mechanical engineering. During the war, President Willard acted as consultant for the U. S. Army and then returned to his duties at the University of Illinois, where since 1920 he has been head of the Department of Mechanical Engineering.

As a consultant on ventilation for the Holland Vehicular Tunnel, under the Hudson River, he assisted the late Clifford M. Holland, chief engineer for whom the tube is named. He supervised the experimental work which was conducted at the University of Illinois and in Pittsburgh. His consulting work has included the proposed Chicago Subway and he has also served the U. S. Bureau of Mines, the Chemical Warfare Service of the Army and the Architect of the Capitol as adviser on heating, ventilating and air conditioning problems.

President Willard is the seventh president of the University of Illinois and in 1928 he was president of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. He is a member of the *American Society of Mechanical Engineers* and Sigma Xi, Tau Beta Pi, Phi Kappa Sigma, Pi Tau Sigma, Sigma Tau and Triangle.

In the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS he has served on many of the Technical Committees, including the Guide Publication Committee, Committee on Ventilation Standards, Committee on Code of Minimum Requirements for the Heating and Ventilation of Buildings, and he has held the offices of President, First Vice-President and Second Vice-President. For a number of years he was a member of the Council and was Chairman of the Executive, Publication, and Program Committees. For several years he was a member of the Committee on Research and since 1931 has been Technical Adviser for Research.

In the field of research, President Willard is known throughout the world and his best known work has been in the field of Heat Transmission, Steam Flow, and Warm Air Heating. He is the author of numerous bulletins, circulars and technical papers published by the University of Illinois and the technical press and is co-author with L. A. Harding of two volumes on mechanical equipment of buildings.

At the time of his appointment as President of the University of Illinois, Walter W. Williams, President of the Board of Trustees, issued the following statement:

"In the election of the new President of the University of Illinois, the members of the Board feel that they have named a man who is not only qualified to become the head of this great State University, but one whose choice will meet with the approval of all who have the affairs of the institution at heart."

This sentiment will be echoed by members of the A. S. H. V. E. and President Willard's host of friends in the engineering field who wish him well and congratulate the University on their choice of a distinguished engineer for President.

Local Chapter Reports

Ontario

March 5, 1934. Forty-four members and guests of the Ontario Chapter were present at the regular March meeting held at the Royal York Hotel. Following dinner, Pres. W. P. Boddington announced the following appointments to the Nominating Committee: W. G. F. Sheppard, chairman, A. S. Leitch and E. T. Flanagan.

President Boddington stated that W. R. Blackhall was representative of the Ontario Chapter on the Coordinating Committee of the Engineering and Technical Societies of Toronto. Colonel Smythe, who has been an active organizer of this movement among engineers in Toronto, was called upon and outlined the following advantages which this action would make possible; to bring influence on municipal problems; to investigate the best method of inviting one group of engineers to technical meetings of other groups; to consolidate all groups so that they will have greater influence and receive more recognition; to secure official assistance in eliminating the troughs and peaks in building construction over a period of time.

H. H. Angus, the principal speaker of the evening, was presented to the meeting and gave a very interesting paper on Engineering Problems in the Design of a Hospital. Following this address there was a considerable amount of discussion, which proved to be of unusual merit.

A vote of thanks was extended by O. L. Maddux before the meeting adjourned, according to Secy. H. R. Roth.

Pacific Northwest

February 8, 1934. At this meeting the Century Brewing Co., Seattle, Wash., were hosts to the Pacific Northwest Chapter, when members of the Society and their friends were conducted through the company's buildings in two groups.

The tour started in the boiler plant and continued through the government metering room and the bottling house. Next the members were shown the boiling processes, during which the beer is boiled four times before being pumped to the cool ship and it was noted that all of the pipes and equipment carrying beer were made of copper. In the cool ship the beer is cooled to approximately 180 deg by washed air and then passed over the Baudelot cooler and into the chip cellar. After being stored for 8 weeks, it is filtered twice and either bottled or put in kegs. The kegs used are made of oak and are pitched inside by the use of high pressure steam.

After the inspection trip the members were served with beer and a Dutch lunch, which, according to D. C. Griffith, will undoubtedly help to make the trip pleasantly remembered by all.

Pittsburgh

January 15, 1934. Pres. P. A. Edwards called to order the meeting of the Pittsburgh Chapter held in the auditorium of the U. S. Bureau of Mines, at which 55 members and guests were present.

The minutes of the preceding annual meeting on December 11, 1933, were read and approved.

Appointment of the following committees selected to serve during 1934 was announced:

Program-Publicity: R. B. Stanger, *chairman;* C. M. Humphreys and L. S. Maehling.
Membership: T. F. Rockwell, *chairman;* J. F. S. Collins, Jr., H. L. Moore and M. A. Benson.
Attendance: E. H. Riesmeyer, *chairman;* H. B. Orr and C. W. Wheeler.

After a discussion of the International Heating and Ventilating Exposition and the 40th Annual Meeting of the Society, the meeting was turned over to the Program Committee.

President Edwards presented John F. Lamb, who spoke on Providing Comfort with Unit Air Conditioners. Mr. Lamb, with his wide experience, ably described the many phases, sides and angles of air conditioning work, and his illustrated talk included consideration of the engineering problems involved in using the unit air conditioner to provide satisfactory conditions in various kinds of occupied space. Mr. Lamb also covered the design problems of the unit air conditioner.

The talk was exceptionally well received by the members and guests present, who engaged in a long and earnest discussion at its conclusion.

After a rising vote of thanks had been tendered Mr. Lamb, the meeting was adjourned at 10:15 p. m.

Western Michigan

February 12, 1934. The February meeting of the Chapter was held at the Hotel Rowe, Grand Rapids, with a good attendance of members and guests. Following dinner, the business session was conducted by Vice-Pres. S. H. Downs, who also gave a very interesting resumé of the Society's Annual Meeting in New York.

The speaker of the evening was L. B. Miller, Minneapolis, Minn., whose subject was Automatic Control for Air Conditioning Apparatus. Guided by a large comfort chart and with the aid of an actual working model set-up of the controls, Mr. Miller presented the problem and demonstrated how its solution was accomplished.

Since the comfort zone covers a large area of coordinates on the chart, a change of one of the axes may or may not call for a change of the other axis. The correlation of these to produce the desirable effective temperatures is the problem. Since the

Summer Meeting in the Pocono Mountains

During the Society's recent Annual Banquet in New York, Merrill F. Blankin of Philadelphia extended a very cordial invitation to all to attend the Semi-Annual Meeting 1934, which will be held at The Inn, Buck Hill Falls, Pa., June 20-22. Philadelphia Chapter is to be host for the occasion and this meeting will be a joint gathering with the *American Society of Refrigerating Engineers,* whose headquarters will be at Skytop.

The Committee on Arrangements has been appointed with the following members: R. C. Bolsinger, *General Chairman;* W. P. Culbert, *Vice-Chairman;* W. R. Eichberg, *Chairman, Entertainment and Registration Committee;* J. H. Hucker, *Chairman, Publicity and Transportation Committee;* W. F. Smith, *Chairman, Finance Committee;* M. F. Blankin, *Chairman, Banquet Committee.*

It is planned to have three technical sessions, one of which will be a joint meeting with the *A. S. R. E.* members.

Invitations have been sent to the Societies' local chapters by the Philadelphia Committee so that preparations can be made by their members to enjoy a vacation and meeting in the heart of the Pocono Mountains of Eastern Pennsylvania.

factors affecting the results of the conditioning system are outdoor temperature, indoor temperature and indoor relative humidity, the controlling instruments are located so as to be affected by these varying influences. These are so grouped together through electric wiring to a common instrument relay as to establish an electrical balance under optimum effective temperature range, by means of relays to the dampers, valves, compressor, etc. A variation of any of the influencing factors, which brings the effective temperature outside the comfort zone, unbalances the mechanism so that that particular portion of the conditioning equipment, which may adjust that influence to bring the space back into the comfort zone, is so adjusted.

The talk and demonstration was presented in an instructive and interesting manner, as was indicated by the responsive discussion that followed.

January 8, 1934. Following dinner at the Hotel Rowe, Grand Rapids, Pres. K. L. Ziesse conducted the business session of the Western Michigan Chapter and introduced the speaker.

H. E. Paetz, president of the Michigan Chapter, discussed the Decalorator, which is a coordinated assembly of apparatus to provide steam jet refrigeration. This study was of special interest to those whose activity is in the fields of steam power, process and heating work. The application of this type of refrigeration for air conditioning brought out much interesting information in the discussion pertaining to balance of year 'round loads, operating costs, etc.

Western New York

February 28, 1934. The February meeting of the Western New York Chapter, held at the Touraine Hotel, was the occasion of the annual dinner dance and reception for officers.

According to Secy. P. S. Hedley there was a very fine attendance and the general impression was prevalent that everyone enjoyed a very pleasant evening.

D. J. Mahoney introduced the newly elected president, J. J. Yager, and presented him with the gavel. President Yager with his customary diplomacy appreciated that this event was neither the time nor place for serious speaking.

Death of James A. Swan

Friends and associates of James A. Swan will regretfully learn of his sudden death on March 8, following a heart attack, at his home in Babylon, L. I., N. Y.

Mr. Swan, who was 45 years old, was a former member of the West Islip Board of Education and was associated with the Revere Copper and Brass Co., New York, N. Y.

Surviving are his widow, Mrs. Jeannie Gregory Swan, and three sons, James A., Jr., John G., and Stetson A. Swan. Funeral services were held at Christ Church, Babylon, on Saturday, March 10, at 3 o'clock.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 25 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by April 16, 1934, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ANDES, WILLIAM, Secy.-Treas., The Andico Co., Cleveland, Ohio.	F. A. Kitchen	R. H. Wehrle
	M. B. Wright	G. L. Tuve
BENSE, WILLIAM M., Student, Pratt Institute, Brooklyn, N. Y.	R. B. Dale (*A.S.M.E.*)	K. E. Quier (*A.S.M.E.*)
	J. W. Hunter (*Non-Member*)	J. L. Eqilernd (*Non-Member*)
COCHRAN, L. H., Dist. Mgr., American Blower Corp., San Francisco, Calif.	H. S. Haley	J. H. O'Brien
	W. E. Leland	F. R. Still
DEAN, FRANK J., JR., Graduate Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt	J. H. Van Alsburg
	G. E. Olson	L. R. Chase
FILLO, FRANK B., Mgr., Minneapolis-Honeywell Reg. Co., St. Louis, Mo.	D. J. Fagin	Paul Sodemann
	J. W. Cooper	A. L. Walters
GILES, ALFRED F., Vent. Engr., H. H. Robertson Co., Pittsburgh, Pa.	H. E. Digby	E. C. Evans
	F. C. Houghten	L. B. Pittock
GLASS, WILLIAM, Mgr., Partridge-Halliday, Ltd., Winnipeg, Canada.	D. F. Michie	J. H. Leonard
	C. D. Kirk	J. B. Steele
HODGE, W. B., Vice-Pres., Parks-Cramer Co., Charlotte, N. C.	J. I. Lyle	L. L. Lewis
	W. H. Carrier	
HYDE, L. L., Genl. Mgr., M. J. O'Neil, St. Paul, Minn.	Albert Buenger	E. F. Jones
	H. E. Gerrish	W. F. Uhl
KEPLER, DONALD A., Student, Pratt Institute, Brooklyn, N. Y.	R. B. Dale (*A.S.M.E.*)	K. E. Quier (*A.S.M.E.*)
	J. W. Hunter (*Non-Member*)	J. L. Eqilernd (*Non-Member*)
KEYS, L. F., Student, Pratt Institute, Brooklyn, N. Y.	J. W. Hunter (*Non-Member*)	R. C. Knowles (*Non-Member*)
	R. B. Dale (*A.S.M.E.*)	H. L. Cameron (*Non-Member*)